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PROVISIONAL SPECIFICATION
FOR A PROVISIONAL PATENT

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Invention Title: **Improvements To Fluid Devices**

The following statement is a description of this invention

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This invention relates to a variation of reciprocating fluid machines colloquially called "scotch yoke" devices.

Known scotch yoke devices comprise one or more pairs of horizontally opposed pistons reciprocating in respective cylinders. Each piston of a pair is rigidly attached to the other so the pair of pistons move as a single unit. The pistons reciprocate along parallel axes which may be coaxial or which may be offset. A crank is provided centrally of the pair of pistons with an offset mounted in a slider. The slider in turn is mounted in the piston assembly between opposing sliding surfaces, which extend perpendicular to the axes of the pistons. The slider is thus constrained to move perpendicular to the piston axes and so, as the crank rotates, the pistons are caused to reciprocate along the piston axis, with a true sinusoidal motion. In certain circumstances the provision of a true sinusoidal motion is preferable to the quasi- sinusoidal motion provided by a crank and connecting rod arrangement found in most internal combustion engines or pumps. However such devices have certain drawbacks. Neither the slider, which reciprocates in a vertical place, nor the pistons, can be dynamically balanced by a rotating mass. Whilst this can be partially compensated for in a multi-pair device, this still leaves rocking couples.

Further in the conventional arrangement the slider slides between a single pair of opposed surfaces which lie on either side of the big end bearing. The pistons must be arranged along parallel axes and the distance between the sliding surfaces of the slider and the guide surfaces of the pistons must be larger than the diameter of the big end on the crank.

The present invention aims to at least ameliorate some of the disadvantages of the prior art and, in preferred forms, provides devices in which paired pistons are not rigidly connected together, are not necessarily coaxial and in which better dynamic balancing is achieved. The invention also allows use of uneven numbers of pistons mounted on a single big end bearing pin.

In its broadest form the invention de-couples the pistons from each other and provides each piston with its own pair or group of sliding surfaces and its own slider. The sliding surfaces for each piston do not lie on either side of the big end but are positioned remote from the big end. The sliding surfaces may be compound surfaces. This decoupling means that each piston is not relying on the coupling with the other piston or pistons to move in both directions and allows the pistons to move along separate axes and at different phases to all other pistons.

Whilst pistons may be interconnected via a common linkage which carries the various sliding surfaces, the pistons are not rigidly connected together. Thus a V-configuration may be achieved with a pair of pistons or a 120° layout with three pistons, for instance.

In one broad form the invention provides a fluid engine or pump, which includes:

a crank mechanism including a big end bearing which orbits about a main axis;

connecting means rotatably mounted on the big end bearing;

at least two pistons, each mounted for reciprocal motion in respective cylinders along a respective piston axis, each piston including guide means which each engage a respective engagement means on the connecting means, said guide surfaces both being disposed on the same side of the big end bearing.

Preferably, the guide means comprise surfaces which extend substantially perpendicular to the respective piston axis. However, the guide surfaces may extend at other than 90° to the respective piston axis. The guide surfaces may deviate from the perpendicular by up to 5° either way. The engagement means may be two or more parallel linear surfaces which correspond and slide relative to the guide surfaces. Alternatively, the engagement means may include two or more roller bearings or the like.

The linear parallel opposed guide surfaces may be located on the connecting means and the engagement means may be mounted on the piston. In preferred forms there are two or three pistons mounted on slider means on each big end bearing. The pistons may be arranged at equal angles about the main axis if desired.

The guide means may be integral with the piston or may be located on a separate structure attached to the piston. Where a separate structure is provided, it may be pivotably mounted to the piston, preferably using a gudgeon pin arrangement. This allows one to use conventional pistons with connecting rods incorporating the guide means.

The crankshaft may be fixed relative to the cylinders or may be movable so as to alter the compression ratio and/ or the timing of the pistons in the cylinders. In a V configuration, movement of the crankshaft along the bisector of the included

angle between the cylinders results in a change in compression ratio without any change in phase. An alternate arrangement provides for the crankshaft axis to rotate about a distant axis, so raising or lowering the crankshaft. These arrangements may be used with a single piston engine. Movement of the crank may be in any direction.

When two pistons per big end bearing are utilised the pistons may be arranged in a V-configuration. The V-configuration may be at any angle, such as 90°, 60°, 72° or any other desired angle. The number of pistons per big end bearing is only constrained by physical size limitations. Each big end bearing may have a single connecting means upon which multiple pistons are mounted or there may be a multiple connecting means mounted on each big end bearing with each connecting means having an associated piston mounted upon it.

When multiple pistons are mounted to one big end bearing, they may be located the same distance from the main axis or different pistons may be at different distances from the main axis.

Whilst the guide means and complimentary engagement means are preferably simple planar surfaces in cross section, other configurations are possible, to provide additional locating surfaces perpendicular to the line of the guide means.

The invention, in another broad form, also provides a fluid engine or pump, which includes:

a crank mechanism including a big end bearing which orbits about a main axis;

connecting means rotatably mounted on the big end bearing;

at least one piston mounted for reciprocal motion in a cylinder along a piston axis, the at least one piston engaging an engagement means on the connecting means whereby the connecting means may have non-rotary movement relative to the at least one piston; and,

stabilising means engaging the connecting means to limit the connecting means to a single orientation as it orbits the main axis.

The stabilising means may include the engagement of the connecting means with the at least one piston. The stabilising means may include a separate linkage pivotably mounted to both the connecting means and the crankcase.

The crank mechanism may be a simple crank with an offset big end bearing or it may be a compound mechanism which provides for other than simple circular motion of the big end bearing at a constant angular velocity. Examples of compound crank mechanisms are disclosed in PCT International Patent Application Nos. PCT/AU97/00030 and PCT/AU98/00287, the disclosures of which are incorporated herein.

The invention, in another broad form, also provides a fluid device, which includes:

a crank mechanism including a big end bearing which orbits about a main axis;

connecting means rotatably mounted on the big end bearing;

at least one piston mounted for reciprocal motion in a respective cylinder along a piston axis, the at least one piston engaging engagement means on the connecting means;

said main axis of the crank mechanism movable along at least one path relative to said cylinder or cylinders and said engagement means configured such that said at least one piston is neither substantially retarded or advanced.

Where the device includes pistons arranged in a V configuration the main axis of the crank mechanism preferably moves along a linear path which bisects the included angle of the V. Alternatively, the main axis of the crank mechanism may move along an arc.

The invention, in another broad form, also provides a fluid device, which includes:

a crank mechanism including a big end bearing which orbits about a main axis, the big end bearing having a big end axis;

connecting means rotatably mounted on the big end bearing for rotation about the big end axis;

at least one piston mounted for reciprocal motion in a respective cylinder along a piston axis, the at least one piston engaging engagement means on the connecting means;

said connecting means having a centre of mass located on or adjacent to the big end axis.

Preferably the crank includes a counter weight which substantially and/or dynamically balances the mass of the connecting means relative to the crank axis.

The invention, in another broad form, also provides a fluid device, which includes:

a crank mechanism including a big end bearing which orbits about a main axis, the big end bearing having a big end axis;

connecting means rotatably mounted on the big end bearing for rotation about the big end axis;

at least one piston mounted for reciprocal motion in a respective cylinder along a piston axis, the at least one piston engaging engagement means on the connecting means; wherein

the effective centre of mass of the crank mechanism, the connecting means and the at least one piston remains stationary or substantially stationary relative to the crank axis as the crank rotates.

The invention, in another broad form, also provides a fluid device, which includes:

a crank mechanism including a big end bearing which orbits about a main axis, the big end bearing having a big end axis;

connecting means rotatably mounted on the big end bearing for rotation about the big end axis;

at least one pair of non opposed pistons, each piston being mounted for reciprocal motion in a respective cylinder along a respective piston axis, each piston engaging engagement means on the connecting means; wherein

the configuration of the connecting means and the engagement means is such that the motion of each piston is simple harmonic motion.

The invention, in another broad form, also provides a fluid device, which includes:

a crank mechanism including a big end bearing which orbits about a main axis, the big end bearing having a big end axis;

connecting means rotatably mounted on the big end bearing for rotation about the big end axis;

at least one pair of non opposed pistons, each piston being mounted for reciprocal motion in a respective cylinder along a respective piston axis, each piston engaging engagement means on the connecting means; wherein,

each pair of pistons has a mass and motion equivalent to a single body orbiting in an ellipse.

Preferably the ellipse is a circle.

Preferably the motion of each of the pistons is simple harmonic motion.

The invention, in another broad form, also provides a fluid device, which includes:

a crank mechanism including a big end bearing which orbits about a main axis, the big end bearing having a big end axis;

connecting means rotatably mounted on the big end bearing for rotation about the big end axis;

at least one pair of pistons, each piston being mounted for reciprocal motion in a respective cylinder along a respective piston axis, the piston axes of each pair being at 90° to each other, each piston engaging engagement means on the connecting means; wherein

each pair of pistons has a mass and motion equivalent to a single body orbiting in an ellipse;

the centre of mass of the connecting means is located on or adjacent the big end axis; and,

the crank includes a counter weight located generally diametrically opposite the big end and a centre of mass remote from the crank axis, the counter weight including the equivalent of:

a first mass to statically and/or dynamically balance all or part of the mass of the big end bearing relative to the crank axis;

a second mass to statically and/or dynamically balance all or part of the mass of the connecting means relative to the crank axis; and,

a respective third mass to statically and/or dynamically balance all or part of the mass of each pair of pistons relative to the crank axis

Preferably the ellipse is a circle and the third mass preferably statically and/or dynamically balances the mass of the pistons.

Where the ellipse is not a circle, the third mass may balance the mass of the pistons in a first direction. The first direction is preferably parallel or perpendicular to a bisector of the axes of each pair of pistons.

In all forms of the invention the connecting means may have non-rotary motion relative to the piston. Preferably there is no rotary motion whatsoever, except as allowed by clearances.

The invention also provides, in one broad form, a reciprocating piston device having:

at least one piston assembly reciprocating along a respective piston axis;

a crank rotating about a crank axis having a drive member offset from the crank axis;

at least one intermediate member located between the drive member and the piston for transferring motion of the drive member to the piston assembly.

The device may have each piston assembly having two surfaces with the offset drive member bearing on one surface and an intermediate member bearing on the other surface.

The device may have a single intermediate member which bears on both surfaces or it may have a two intermediate members each of which bears on one of the respective surfaces.

Each piston assembly may have one piston or it may have two or more pistons. Where two pistons per assembly are provided, preferably the at least one intermediate member is located between the pistons.

The drive member is preferably a circular cam having its centre offset from the crank axis.

The device may have two or more piston assemblies for each drive member.

Where two or more pistons assemblies for each member are provided, they may reciprocate along piston axes extending at any angle to each other. Preferably there are two piston assemblies per offset member extending at 90° to each other.

Where two piston assemblies extending at 90° to each other are provided, preferably there are provided two intermediate members, each of which engages both piston assemblies.

Preferably the device includes stabilising means which engage the respective piston assembly to limit or substantially prevent movement of the respective piston assembly transversely of the respective piston axis.

The stabilising means may be mounted on or be integral with the crankcase. The stabilising means may include a slot or elongate recess in the piston assembly. The slot or recess may engage a member on the crankcase or it may engage the crankshaft itself, directly or indirectly. If the slot or recess engages the crankshaft, preferably there is provided a follower intermediate the crankshaft and the slot or recess.

The device may include subsidiary one or more fluid pumps for pumping lubricating fluid of the like to specific locations of the device. Each fluid pump may be mounted on the crankcase, the crank assembly, the drive mechanism mounted on the crank or on a piston assembly. Any combination or these positions is possible. The fluid pump may be used to cause secondary movement of the piston assembly relative to the drive member and/or the crank.

The crank may include a secondary drive member which engages the at least one intermediate member or the piston assembly. The secondary drive member may contact either continuously or intermittently as the crank rotates. The secondary drive member may drive one or more of the aforesaid secondary fluid pumps.

The invention shall be better understood from the following, non-limiting description of preferred forms of the invention, in which:

Figure 1 is a cross-sectional view of a fluid machine according to the invention.

Figure 2 is a partial cutaway perspective view of the Figure 1 device.

Figure 3 is a perspective view of a three piston fluid machine according to the invention.

Figure 4 shows an end view of a third embodiment of the invention.

Figure 5 shows a partial cutaway perspective view of a fourth embodiment of the invention.

Figure 6 shows an end view of a connecting device of the fig 5 device.

Figure 7 shows a perspective view of the fig 6 device.

Figure 8 shows an end view of a variation of the Figure 1 embodiment.

Figure 9 shows a perspective view of a fifth embodiment of the invention.

Figure 10 shows an end view of the Figure 9 embodiment.

Figure 11 shows an end view of a sixth embodiment of the invention.

Figure 12 shows an end view of a seventh embodiment of the invention.

Figure 13 shows an end view of a eighth embodiment of the invention.

Figures 14 to 28 and figures 31 to 48 show various configurations of the guide surfaces of the invention.

Figures 29 and 30 and figures 49 to 66 show further embodiments of the invention.

Figures 67 to 80 show further embodiments of the invention.

Referring to Figures 1 and 2 there is shown a fluid device 10 which includes a crank 12 mounted for rotation about a crank axis 14. The crank 12 has an offset bearing pin 16, radially distant from the axis 14. Thus as the crank 12 rotates about axis 14, pin 16 will describe a circular orbit around axis 14.

Rotatably mounted on bearing pin 16 is a slider 18. The slider has two tongues 20, 22.

The slider 18 extends generally perpendicular to the axis 14 whilst the tongues extend generally parallel to the axis 14. As best seen in Figure 2 the sliding surfaces extend axially on either side of the main portion 24 of the slider and so form a T-shaped construction.

Each of the tongues 20, 22 engages in a T-shaped slot 30 of a respective piston 32. Each piston is mounted in a cylinder 34 and constrained for linear movement along respective cylinder axis 36. The slot 30 preferably extends substantially perpendicular to the cylinder axis 36 and extends diametrically across the centre of the piston. Both ends of the slot 30 are open. The slider can thus move sideways relative to the piston but must move axially with the piston. Where the slot 30 does not extend at 90° to the piston axis, sideways movement of the tongue relative to the piston will cause axial motion of the piston. This enables one to control the motion of the piston beyond a pure sinusoidal motion.

The piston is constrained to move along its piston axis and as the crank 12 rotates the slider 18 member rotates about the crank axis 14. The motion of each tongue has a component parallel to the respective piston axis and a component perpendicular to the respective piston axis. Thus, the pistons reciprocate in their respective cylinders with the tongues sliding sideways in their respective slots 30. The combination of the linear movement of the piston and the tongue in the slot maintains the slider member 18 in a constant orientation as the crank rotates, irrespective of other pistons. In the embodiment of Figure 1, there are provided two pistons at 90° to each other, but since the slider 18 maintains its orientation as it orbits the crank axis, the angle between the pistons may be other than 90°. Similarly more pistons may be added.

Figure 3 shows a perspective view of a three piston device. For clarity the cylinder and crank cast assemblies are omitted. As can be seen, the device 110 includes a crank 112 with a bearing pin 116 extending between webs 117. Three pistons are arranged equally about the crank at 120° to each other. Mounted on the bearing pin is a triple tongue device 118. This device may be a unitary structure or it may comprise three separate components mounted on the pin 116. As seen, each piston is provided with a T-shaped slot 130 into which the respective tongue 120 engages. The pistons are axially offset but, if desired, they may be in a common plane.

Because each of the pistons is decoupled from any other piston, the orientation and position of the pistons may be chosen as desired. There is no need for the piston axes to extend radially from the crank axis. The piston axes may extend radially from an axis, but this axis may be remote from the crank axis. The piston axes may be parallel and spaced from each other on either side of the crank axis.

Figure 4 shows a fluid device 50 having a crank 52 rotating about crank axis 54. A slider mechanism 56 is maintained on a bearing pin 56 and has two arms 58, 60 which extend horizontally and engage in slots 62, 64 respectively of pistons 66, 68. Each of the pistons 66, 68 reciprocates in a dual chambered cylinder 70, 72. The cylinders 70, 72 are closed at both ends and thus combustion chambers 74 are defined between the pistons and the ends of the cylinders.

Rotation of the crank 52 causes the pistons to reciprocate vertically within the cylinders with the arms moving sideways relative to the pistons.

Referring to figures 5 to 7 there is show a reciprocating piston device 210 having two pistons 230 reciprocating in respective cylinders 234 at 90° to each other. A connecting device 218 connects the two pistons to big end pin 216 of crankshaft 212 via tongues 220 and slots 230 in the pistons 232. The connecting device 218 has two webs 240, one for each piston, which are offset axially relative to each other. This allows the pistons 232 to overlap each other and so be brought closer to the crank axis 214. Lubrication ducts 242 are provided to supply pressurised oil from the big end pin 216 to the sliding surfaces of the tongues 220and slots 230.

The connecting device 218 includes a counter weight 244 extends downwardly on the opposite side of the big end pin 216, bisecting the angle between the two webs 240. This counter weight 244 is sized so that the centre of inertia and preferably also the centre of mass of the connecting device 218 lies on the big end axis 246. It will be appreciated that when the pistons are spaced equally about the crank axis 214 that the webs 240 will balance each other and a separate counter weight may not be needed.

As the connecting device orbits the crank axis 214 no rotational forces are generated relative to the big end axis 246, which would cause the connecting device to attempt to rotate about the big end and which would need counter turning forces to be generated at the slot 230 / tongue 220 interface. In addition, since the centre of inertia of the connecting device remains on the big end axis 246, it is a relatively simple matter of adding an appropriate amount of mass to the counter weight 248 on the crank 212 diametrically opposite the big end axis

246 to provide a dynamically balanced crankshaft/connecting device combination. It will be appreciated that for other piston arrangements that so long as the connecting device's centre of inertia lies on the big end axis 246, then it may be dynamically balanced.

This leaves the reciprocating mass of the pistons. The velocity of the pistons follows a pure sinusoidal path and in combination the two pistons are the equivalent of a single rotating mass. This may be balanced by adding an appropriate mass to the crankshaft, thereby resulting in a dynamically balanced device. For a V twin configuration, a single piston mass is added to the back of the crankshaft. For a four piston star configuration, two piston masses are added to the crank counter weight.

Referring to figure 8, there is show a fluid device 50 which is a variation on the figure 1 embodiment. For clarity the same numbers are used for the same components. The combination of the piston 32 being limited to linear motion along the piston axis 36 and the respective tongue 20 being limited to linear motion relative to the piston 32 theoretically prevents any rotation of the connecting means 18 relative to the piston 32. However, due to the need for manufacturing tolerances, there will inevitably be some free-play and hence turning of the connecting means 18 relative to the pistons 32. This in turn will generate turning forces at the interfaces of the tongues 20 with the slots 30. To alleviate this, the device in figure 8 is provided with a linkage 40. One end of this linkage 40 is pivotably connected to the connecting means 18 at 42 and its other end is pivotably connected to the crankcase (not shown) at 44. The linkage 40, connecting means 18, crankshaft 12 and crankcase thus for a four bar linkage. The distance between the two pivot points 42, 44 is the same as the separation of the crank axis 14 from the big end axis 46. Thus, irrespective of the restriction imposed by the engagement of the connecting means 18 with the pistons 32, the connecting means is constrained to orbit about crank axis 14 without changing its orientation.

Referring to figures 9 & 10 there is shown a twin cylinder fluid device 60 having pistons 62 reciprocating in cylinders 64. The pistons 62 are each provided with a gudgeon pin 66 mounted in a bearing 68 on the respective piston. Mounted on the gudgeon pin 66 is a connecting rod 70. However, the connecting rod 70 does not mount on the big end of the crankshaft 12, but on the connecting means 18. The lower end 72 of the respective connecting rod 70 is provided with a T-shaped slot 74 which receives the T-shaped tongues 20 of the connecting means 18.

Whilst the connecting rod 70 is free to rotate about the gudgeon pin 66 relative to the piston, the combination of the planar mating surfaces of the slots 74 and tongues 20 prevents any pivoting and so the connecting rod 70 and connecting means 18 move as a single unit. Whilst this may appear to introduce unnecessary complication to the structure, it does allow one to use conventional pistons.

Referring to figure 11, there is shown a twin cylinder fluid device 80 with twin pistons 82 mounted on connecting means 18 in cylinders 84. The connecting means 18 is mounted on a crankshaft 12, but the axis 14 of the crankshaft is not fixed relative to the cylinders 84. Instead, the crankshaft 12, and with it connecting means 18 and pistons 82 may be moved upwards or downwards, as indicated by arrows 86. The vertical movement of the crankshaft 12 raises the pistons in the cylinders 84 and thus provides the ability to vary the compression ratio on the fly. Movement of the crankshaft 12 does not effect the timing of the pistons in the cylinders 84 relative to the crankshaft 12 or to each other. This is in contrast to conventional V engines which if provided with movable cranks, causes the timing of the pistons to vary, with one piston being advanced and the other retarded.

Vertical movement of the crankshaft 12 may be achieved utilising conventional means, such as hydraulic rams or the like.

It will be appreciated that a movable crank may be utilised with a single piston and that the movable crank may be moved along paths other than the bisector in a V-twin engine, for example. The crank may be moved at a, say, 15° to the vertical. This has no effect other than to need more crank movement to achieve the same change in compression ratio.

Figure 12 shows a variation of the figure 11 embodiment, in which the crankshaft 12 is mounted on bearing arms 90. The crank engages a gear 92, which may be connected to a gearbox in the case of an engine. The gear 92 has an axis of rotation 94. The bearing arms 90 are pivotably mounted on the crankcase about axes which are coaxial with the axis 94. The bearing arms may be rotated about the axis 94 by suitable means to raise or lower the crankshaft relative to the cylinders. Whilst this does cause a sideways movement of the crankshaft, and so advancement and retardation of the pistons, this is very slight.

Figure 13 shows a further embodiment of the invention, in which there is a twin cylinder device 100 with pistons 102 reciprocating in cylinders 104. The pistons have connecting rods 106 pivotably mounted on gudgeon pins 108. The lower

end of the connecting rod is provided with two opposed parallel surfaces in which a slider 110 is mounted. The opposite ends of the slider 110 are connected to hydraulically operated rams 112. These rams 112 are incorporated within the connecting means 18 and are selectively supplied with high pressure oil via ducts 114. The rams 112 are thus capable of causing the slider 110 to pivot about its centre 116, to raise or lower relative to the connecting means 118, and hence relative to the cylinder, or a combination of both. This causes the piston to rise or fall relative to the respective cylinder and/or for the connecting rod 106 to pivot about gudgeon pin 108, so altering the phase of the piston.

Figures 14 to 29 show a number of variations of the guide surfaces of the piston and the corresponding surfaces on the engagement means.

Figure 14 shows a slider 100 having a Y-shaped engagement surface 102 for engagement with two pairs of surfaces 104, 106 of a single piston.

Figure 15 shows a slider 110 having engagement means 112. This surface 112 is Y-shaped but has surfaces 114, 116 extending from base 118.

Figure 16 shows a slider 120 having engagement means 122. The engagement means in cross section is T-shaped with two arms 124, 126. These arms 124, 126, in cross section, form a curved upper surface 128.

Figure 17 shows a slider 130 having an arrow-headed engagement means 132. The engagement means 132 has two downwardly extending and diverging arms 134 which are engaged by the piston.

Figure 18 shows a W-shaped engagement means 140.

Figure 19 shows a T-shaped engagement means 150 but the upper and lower surfaces 152, 154 of the arms 156 are provided with V-shaped grooves 158, in which V-shaped protrusions 160 extend. The V-shaped grooves 158 and protrusions 160 may be located on the other of the piston and engagement means.

Figure 20 shows a T-shaped engagement means 170 having an upper surface 172 with a slot 172 located centrally therein. The corresponding surface 174 of the piston includes a rectangular shaped protrusion 176 which extends into the slot.

Figure 21 shows a T-shaped engagement means 190 having a semi-circular protrusion 192 located centrally on the upper surface 194. The protrusion 192 need not be located centrally and there may be additional protrusions located on one or both sides of the centre of engagement means, either on the upper surface 194, the lower surfaces 196, 198, or both.

The device of Figure 22 is similar to that of Figure 17 except that the upper engagement surface 180 of the piston is not continuous but is provided with an opening182.

Figure 23 shows a T-shaped engagement means.

Figure 24 shows a T-shaped engagement means 210 having arms 212 and 214. The side surfaces 216, 218 of the arms are curved, so the width between the surfaces 216 and 218 is greater at the centre of the engagement means than at either end. It will be appreciated that the width of the corresponding slot in the piston will need to be at least as wide as the widest part of the two arms.

Figure 25 shows an engagement means 220 which is T-shaped but in which the arms 222, 224 converge in the longitudinal direction, so as to form a triangular shaped upper surface.

Figure 26 shows a T-shaped engagement means 230 having arms 232 and central leg 234. The leg 234 is provided with linear gears 236, 238 on its two surfaces. These gears 236, 238 may be used to drive, via rotatable gears mounted on the piston, other devices.

Figure 27 shows an end view of the figure 25 embodiment.

Figure 28 shows a T-shaped engagement means 250 having a centrally located linear gear 252 on the upper surface 254. As with the figure 26 device, this gear may be used to drive devices mounted on or in the piston.

Figure 29 shows a V-twin engine 300 having pistons 302, crank 304 and connecting means 306 mounted on the big-end 308 of the crank 304. The pistons are conventional pistons in having a gudgeon pin 310 on which is rotatably mounted a connecting rod 312. However the connecting rods 312 have a slot 314 at their lower end in which the connecting means 306 engages.

The connecting rods each have a sideways extending arm 316 which engages a slider 318 which slides in guides 320 parallel to the respective cylinder axis. The

connecting rod may be integral with the slider 318 or it may be connected by way of a pivotable joint 322, as shown. The joint 322 may be a single axis joint or a ball type joint. In the embodiment shown the arms 316 extend parallel to the slots 314. However they may extend at any angle.

The guides 318 aid in stabilising the respective piston because the tolerances required can result in the piston rotating very slightly in the bore and cause seizing or the like. If very tight tolerances are used, the guides may not be needed. The sliders 318 may be integral with the crank case or may be separate items attached to the crank by way of bolts and the like

The gudgeon pins of the pistons may be at 90° to the crank axis as no rotational movement of the connecting rod relative to the piston will occur. Use of the pistons with gudgeon pins allows one to use "off the shelf" pistons.

Figure 30 shows a schematic layout of a V-twin engine having a primary crank 330, a big end 332 and a connecting means 334 mounted on the big end. Pistons 336 are mounted on the connecting means 334 as in the previous embodiments.

A slave crank, 338 is provided which rotates about an axis 340 parallel to the axis 331 of the primary crank. A link 342 is pivotably mounted on both the connecting means 334 at 344 and the slave crank 338 at 346. The distance of pivot point 346 from the slave axis 340 is the same as that of the big end 332 from the primary axis 331. The slave crank and link 342 thus aid in maintaining the connecting means in a fixed orientation as the primary crank 330 rotates. It will be appreciated that this stabilisation technique may be used with any of the embodiments described herein.

Figure 31 shows an axial cross-section through a big end 350 and a connecting means 352. The connecting means 352 has engagement means 354 which is engaged by engagement means 356 and 358 of two separate pistons (not shown).

Figure 32 shows a similar structure to that of Figure 31 but with a different configuration of the engagement means 360 on the connecting means 362 and the corresponding engagement means 364, 366 of the two pistons.

Figure 33 shows a connecting means 370 having two slots 372, 374 in each of which is engaged a T-shaped engagement means 376, 378. The engagement means 376, 378 may be attached to a single piston or to separate pistons.

Figure 34 shows a connecting means 380 having two slots 382, 384. Each slot has a Z-shape which traps the corresponding engagement means 386, 388.

Figure 35 shows a connecting means 390 having two slots 392, 394 in which are received engagement means 396, 398. Located in the slots are roller bearings 400 to aid movement of the engagement means 396, 398 along the slots 392, 394. It will be appreciated that the bearings 400 will be provided at intervals along the slots.

Figure 36 shows a connecting means 410 in which the piston engagement means 412, 414 surround the connecting means 410 and engage in downwardly opening slots 416, 418.

Figure 37 shows a connecting means 420 having two sideways opening slots 422, 424. Two engagement

Figure 38 shows a connecting means 430 having a T-shaped engagement means 432 having arms 434 and 436 which descend divergently. The upper and lower surfaces 438 and 440 maybe parallel, as in arm 434 or divergent as in arm 436. The piston has a series of opposed roller bearings 442 which engage the upper and lower surfaces. As examples the centre line of the arms maybe at between 35° and 50° to the big end axis.

Figure 39 to 47 show further variations possible of the connection between the connection means and the engagement means of the piston or pistons mounted thereon.

Figure 48 shows a twin piston fluid device 500 having two opposed pistons 502 and 504 reciprocating in cylinders 506 having a common axis. The pistons are rigidly joined by a linkage 507. Two crank shafts are provided 508 and 510 having connecting means 512 mounted on respective big ends 514 and 516. The connecting means passes through a bore 518 in the linkage 507 which constrains the connecting means 512 to slide sideways relative to the linkage. Preferably this motion is perpendicular to the cylinder axis but need not be so.

The two cranks 508 and 510 are preferably linked, such as by gears, so that they rotate together. As they rotate the connecting means describes a sinusoidal vertical motion and so causes the pistons to describe similar motion.

Figure 49 shows a variation of the Figure 48 device and accordingly like parts are numbered the same.

Whilst the connecting means 512 is free to slide sideways relative to the linkage 507, there will be some sideways loading on the linkage. Accordingly guide surfaces 520 and 522 are provided either side of the linkage 505 to prevent sideways motion.

Figure 50 shows a variation of the Figures 48 and 49 devices. In this embodiment four pistons 530 are pivotally connected to a slider 540 by linkages 542. The pistons are arranged in an X configuration. This may be one with equal angles of 90° between each piston path or, as shown, one where the pistons are not equally spaced. Each linkage pivotably connects to the respective piston and slider. The slider 540 is constrained to move vertically by guides 520 and 522 and does so sinusoidally.

Figure 51 shows a device 600 similar to the Figures 48 to 50 devices in having two cranks 602 and 604 rotating together. An X-shaped connecting means 606 is mounted on the big ends 608 and 610 of the two cranks. Four pairs of linked pistons 512, 514, 516 and 518 are each mounted on one of the arms 520 to 524 of the connecting means. Each arm and its associated pair of pistons is the equivalent of the Figure 48 device. The arms 520 to 524 preferably extend at 90° to each other, but this is not essential. Further it is not essential that one of the big ends be mounted to the centre of the connecting means. Preferably the two cranks are located either side of the centre of the connecting means.

Figure 52 shows a further embodiment of the invention which includes an opposed piston device 700 having pistons 702 reciprocating in cylinders 704. The pistons 702 are rigidly joined by linkage 706 and so move together. Mounted between the pistons is a crank 708 which rotates about axis 710. The crank 708 has a circular disk 712 which is offset from the axis 710, having its centre at 714. Thus as the crank rotates the disk oscillates vertically and horizontally. Mounted on the linkage 706 are two followers 716. These followers 716 bear against vertical surfaces 717 of the linkage and may move vertically relative to the linkage 706 but not horizontally. The followers have circular surfaces 720 which engage the circular surface of the disk 712.

Figure 53 shows a single piston device 730. This is substantially the same as the fig 52 device, except one of the pistons has been omitted. The main bearing 732 of the crank 708 has a split member 734. This member 734 is located in a slot 735 of the piston assembly and so aids to stabilise the piston assembly.

Figure 54 shows a twin piston device 800 having two pistons a 90 degrees to each other. However other angles may be used. The pistons 802 and 803 are not in a plane but are staggered axially along the axis 804 of the main crank 806. The main crank 806 has an offset cylinder 808. The first piston 802 has two arms, 810 & 812 extending on either side of the cylinder 808. Mounted on the arms are followers 814 which engage the cylinder 808 and which translate the oscillating motion of the disk 808 into a reciprocating motion parallel to cylinder axis 816. Similarly, the second piston has a similar pair of arms (obscured in the drawing) which hold followers 820. These followers also translate the oscillating motion of the cylinder 808 into reciprocating motion along cylinder axis 822. The device also has guide members 821 which engage the piston to limit sideways motion of the pistons.

Figure 56 shows a four cylinder engine 830 having two pairs of linked piston assemblies 832. Each assembly 832 has a piston 834 at each end rigidly joined to the other by a linkage 838. The pistons 834 of each assembly reciprocate in cylinders 836.

Each linkage 838 has a slot 839 extending in a vertical plane through the linkage. Each slot has parallel vertical end walls 841 and located in each slot is a slider 844, having parallel vertical end walls 846. Each slider is free to move vertically in the respective slot.

A crank 840 extends horizontally through the linkages 838 and the sliders 844. The sliders 844 each have a circular opening 848 through which the crank passes. The crank has a circular cam 842 which has a size corresponding to the opening 848. The cam centre is offset from the crank axis and so as the crank rotates, the cam centre orbits the crank axis. This causes the slider 844 to move vertically and horizontally relative to the crank axis.

Vertical motion of the sliders 844 is "lost" due to the vertical freedom of the sliders relative to the piston assemblies, whilst horizontal motion causes the piston assemblies to oscillate horizontally in a true sinusoidal motion.

This construction has a number of advantages over existing similar systems. The main advantage is that interposing of a slider between the cam 842 and the slot

walls 841 removes application of point loads, which would otherwise occur. Instead the load is transferred over large surfaces from the cam to the slider and from the slider to the slot.

Figure 57 shows a twin piston engine 850 similar to that of figure 56 and accordingly like parts have the same numbers. In this embodiment two sliders 852 are provided, one on each side of the cam 842. Each slider does not contact the other and so each is "floating" relative to the other. In the figure 56 embodiment, if the slider 844 rotates relative to the slot 839, there is a tendency for it to jam in the slot. The use of a "split" slider by using two sliders prevents this occurring. If one of the sliders rotates relative to the slot, then all it does is rotate around the centre of the cam 842.

Figures 58 to 60 show a four piston device 860 having pairs of pistons 862a,b arranged at 90° to each other. Each piston has an extension 864 having surfaces 868 and 870 extending perpendicular to the respective piston axis. The extensions 864 extend to one side of the piston axis, as best seen in figure 59, so that the pistons of each pair may be positioned in a common plane perpendicular to the crank 866.

The crank includes an offset circular cam 872 which engages the four walls 868a,b, 870a,b. As the crank rotates, the cam 872 causes both pistons 862a, b to reciprocate in their respective cylinders, not shown.

Whilst the figure 58 to 60 embodiment uses a cam bearing directly on the end walls, it will be appreciated that the slider construction of the figure 56 or 57 embodiments may be utilised.

Figures 61 and 62 show a variation of the figure 58 to 60 embodiment and so like parts are numbered the same.

Two sliders 880 are interposed between the cam 872 and the end walls 868 and 870. Each slider bears on the inner face 868 of one piston and the outer face 870 of the other piston. As the crank 866 rotates this causes the sliders to move both pistons. It will be appreciated that as a piston moves toward the crank, the slider bearing on the respective end wall 870 will push the piston toward the crank whilst as the piston moves away from the crank, the other slider bearing on the inner wall 868 will push the piston away from the crank. As with the figure 57 embodiment, since each slider only bears on one end wall of each piston, the likelihood of jamming is reduced.

Figure 63 shows a V-twin device 882 similar to that of figures 58 to 60 in which a cam bears directly on end walls of the pistons. As such, like parts have the same numbers. To aid stability of the pistons 862, guides 884 are provided which engage either side of the extensions 864 to prevent sideways motion of the piston relative to the respective piston axis.

Figure 64 shows a V-twin device 890 in which each extension 864 is provided with a longitudinally extending slot 892 through which the crank 866 extends. The slot 892 allows longitudinal motion but not transverse motion. If desired a slider block may be positioned on the crank to engage the slot walls.

Figure 65 shows a hybrid slider arrangement which may be utilised for any of the embodiments described herein. In this embodiment there is provided a crank 900 having an offset circular cam 902. The cam 902 is located in a slot 904 of a single or twin piston unit having end walls 906 and 908. The slot is longer than the diameter of the cam 902 and a slider is located between the cam and the end wall 908. The cam bears directly on the end wall 906.

Figure 65 shows a scotch yoke engine 910 having twin opposed pistons 912. A crank 914 has a big end 916 upon which is mounted a slider structure 918 which slides along guide surfaces 920, 922 as the crank 914 rotates, thereby causing vertical motion of the pistons. This structure comprises two independent pieces 924, 926. These two pieces 924, 926 engage surfaces 920, 922 respectively. The split line between the two pieces 924, 926 runs at about 30°; but may be at any angle.

Figure 66 shows an X configuration scotch yoke engine 930 having two piston assemblies 932, 934. A "split" slider assembly 936 engages the sliding surfaces of the two piston assemblies. The slider assembly comprises two slider pieces 938, 940 both of which engage both piston assemblies. However, each piece only contacts one sliding surface of each piston assembly.

Figure 67 shows a two piece slider assembly 942 mounted on the big end 944 of a crank 946 whilst the assembly is comprised of parts 948, 950, they are rigidly joined together by bolts 952, so the structure acts as a unitary structure.

Figure 68 shows a slider assembly 954 mounted on a big end 956. The assembly has two components 958, 960, each of which bears against one of the sides of a slot of a scotch yoke type engine. Each of the two components has a loop 962 which surrounds the big end and allows the respective component to rotate about

the big end independent of the other. It will be appreciated that the loop may be separate from the body 964 of the component and attached by bolts or the like.

Figure 69 shows a slider assembly 965 comprised of two components 966 mounted on cam 968. Each component 966 engages one side of the guide slot of a piston assembly. Each component 966 in turn is comprised of two parts 968, 970 linked by linkages 972. The linkage 972 may be rigidly attached to each component or pivotably mounted.

Figure 70 shows a detail of one side of a slider assembly in which two parts are pivotably joined at axis 974.

Figure 71 shows one part of a slider assembly having a component 976 which engages an off centre cam 978. The component has a main body 980 and rollers 982 which normally engage surface 984 of the slot and hold the main body just above the surface 984. As the cam rotates, the velocity of the component along the surface changes. The separation of the body from the surface is sufficiently small that at high velocity the body floats on a film of oil and at low velocity it is supported by the rollers.

Figure 72 shows a slider assembly where the slider component 986 engages the cam 988 by way of rollers 990.

Figure 73 shows a multi-part slider assembly 992 having parts 993, 994 engaging on the sides of slot 995. The two parts are joined by parts 996, 997 which closely follow the surface of the cam 998 to aid in maintaining hydrodynamic lubrication of the slider parts on the cam.

Figure 74 shows a slider assembly 1000 having two parts 1002, 1004 on either side of a cam 1006. Linkages 1008 join adjacent ends of the two parts 1002, 1004. The linkages may be rigidly or pivotably attached to the parts.

Figure 75 shows a slider assembly having a link 1010 joining diagonally opposite ends of a two part slider assembly 1012.

Figure 76 shows a crank 1014 having a first, circular cam 1016 which is engaged by slider parts 1018, 1020. A second cam 1022 is located adjacent the first cam or is superimposed on the first cam 1016 and engages a cam follower 1024 mounted on the respective part once per crank revolution.

Figure 77 shows a two part slider assembly with each part 1031 having a sliding surface 1030 engaging on surface 1032. Each part 1031 also carries a roller 1034 which intermittently engages a cam surface 1036 on the piston assembly so as to move the piston assembly further away from the crank.

Figure 78 shows a slide 1040 and piston assembly 1042 with linear sliding surfaces 1044 and 1046 respectively. The piston assembly has cam surfaces 1048 which are engaged by followers 1050. These followers are connected to pistons 1052 on the slider so as to pump out lubricant as needed. It will be appreciated that the cam/follower/pistons may be reversed so the cam surface is on the slider.

Figure 78 shows a crank 1060 having a main, circular cam 1062 which is engaged by slider components 1064. Each slider component has a cam follower 1066. This cam follower is intermittently engaged by a second cam 1068 as the crank rotates.

Figure 79 shows a variation of the figure 78 device in which the cam follower drives a pump 1070 to intermittently drive oil to various bearing regions.

Figure 80 shows a scotch yoke assembly having a unitary slider 1072 mounted on a big end 1074. The slider 1072 has an oil pump 1076 which is intermittently engaged by a cam 1078 located on the crank.

It is to be understood that the various forms of the slider and the engagement means on the sliders may be used with any of the other forms of the invention in any practical combination possible and the various forms are not limited to use with the components shown in the specific figures.

It will be apparent to those skilled in the art that many modifications and variations may be made to the embodiments described herein without departing from the spirit or scope of the invention.

Dated this 9th day of June 1999

MICHAEL JOHN RAFFAELE & PETER ROBERT RAFFAELE

by their Patent Attorneys

CHRYSILIOU MOORE MARTIN

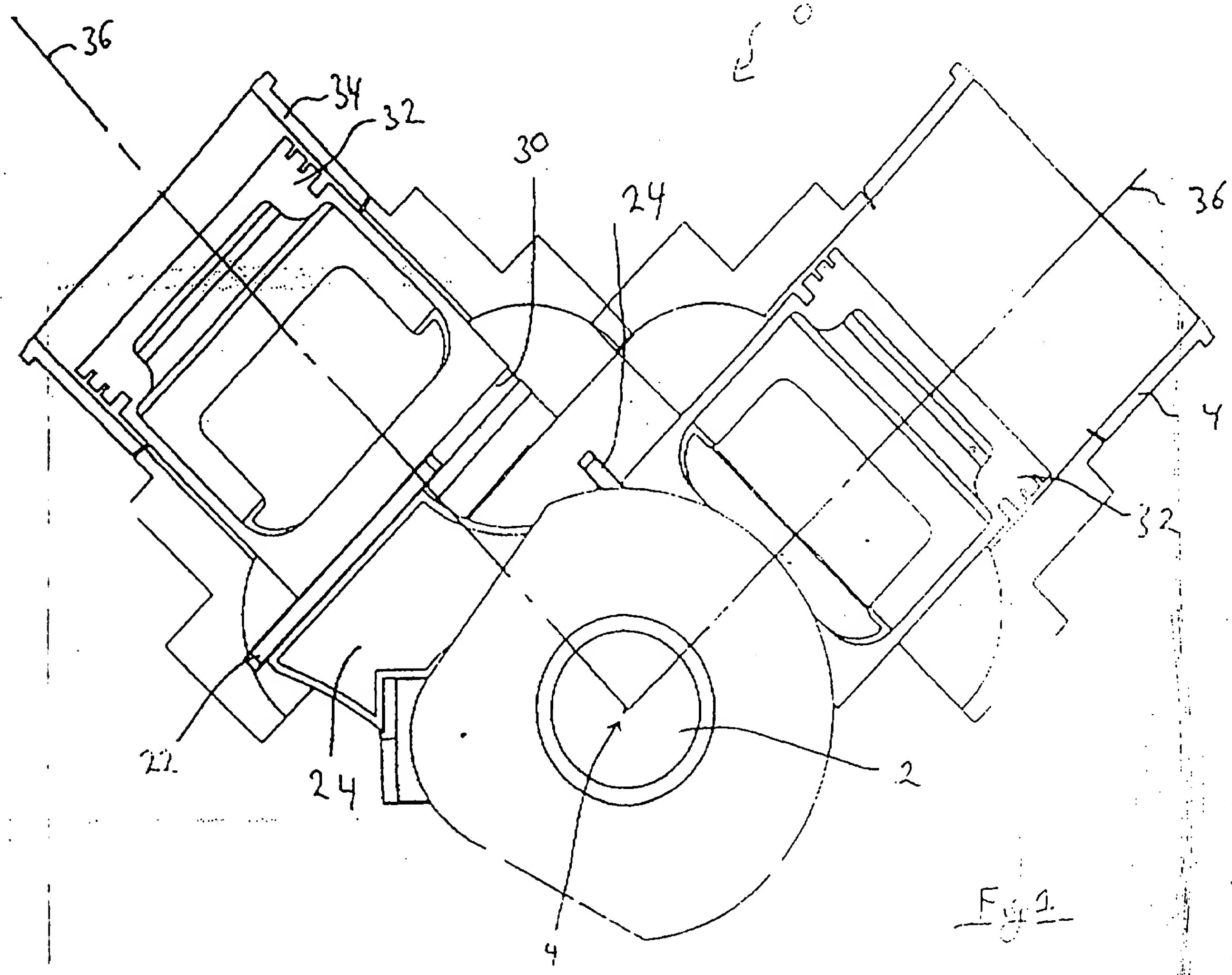


Fig 1

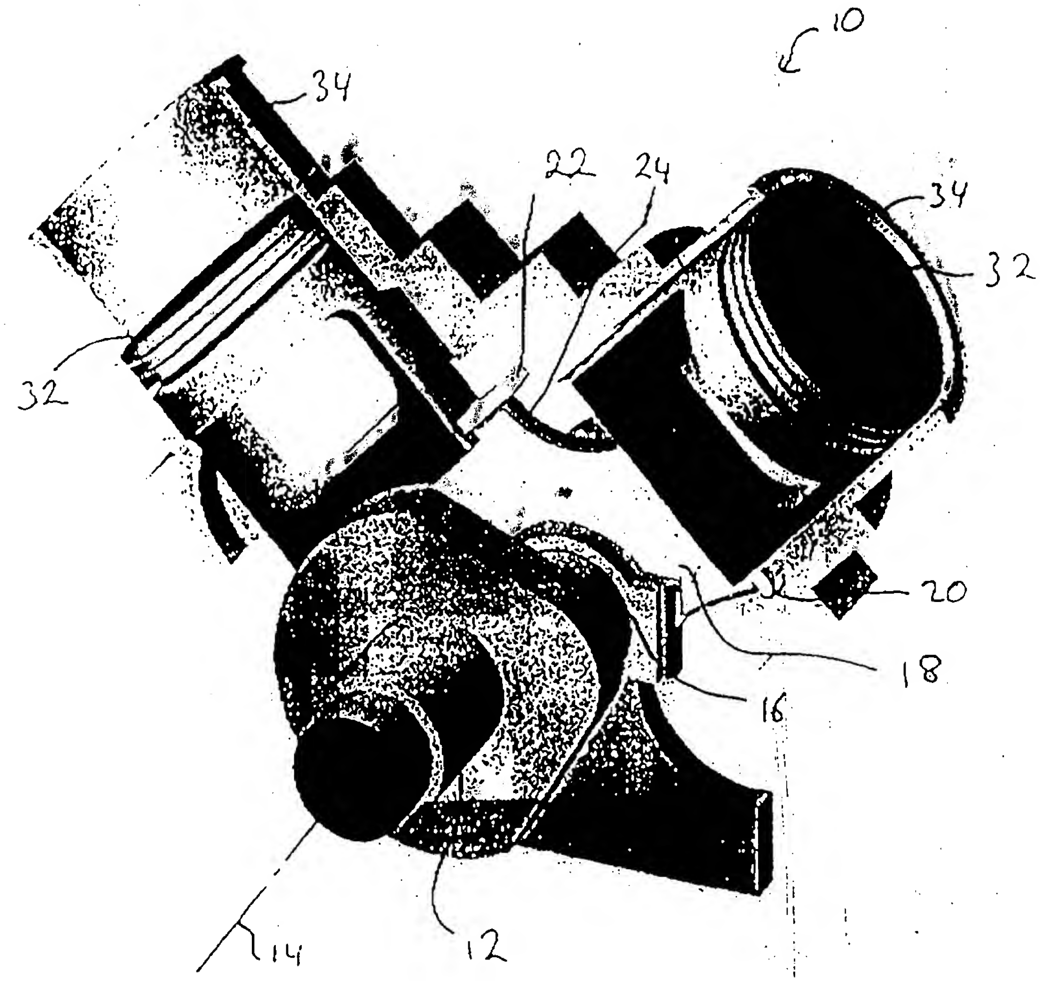


Fig 2

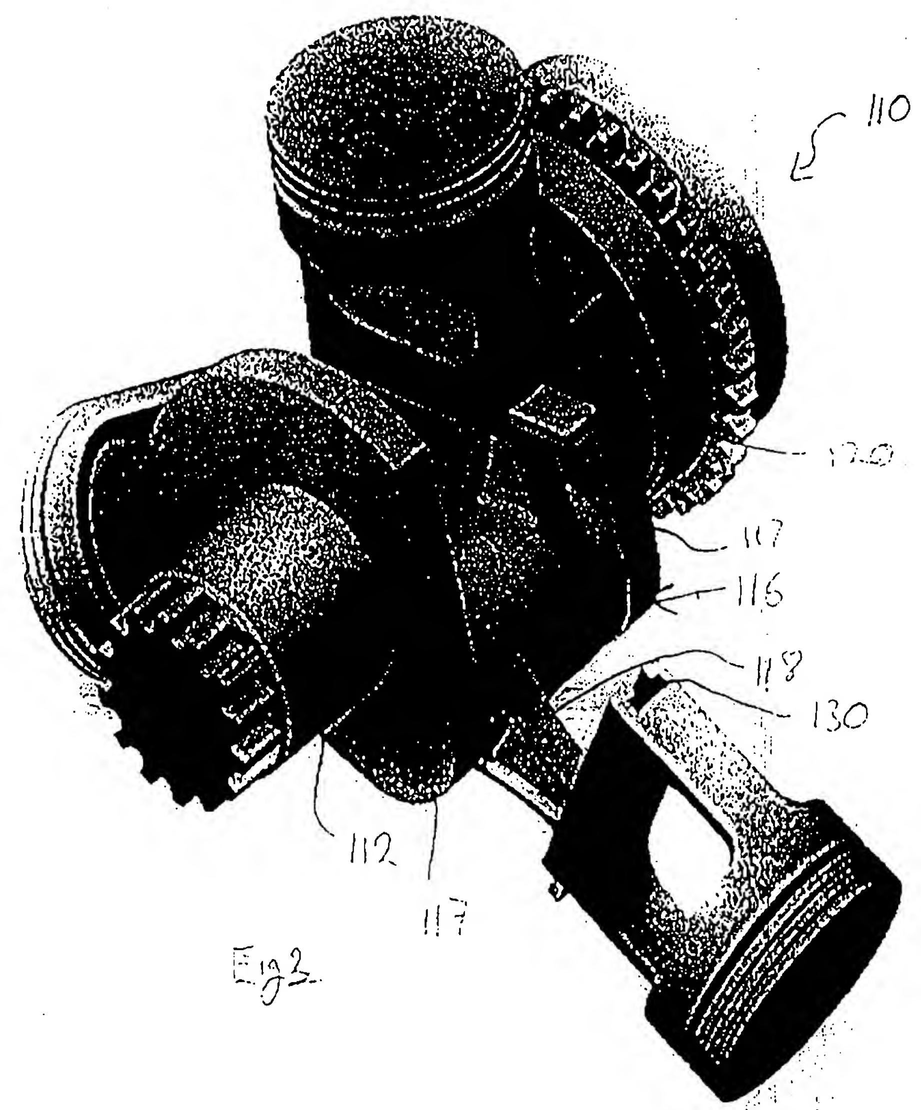
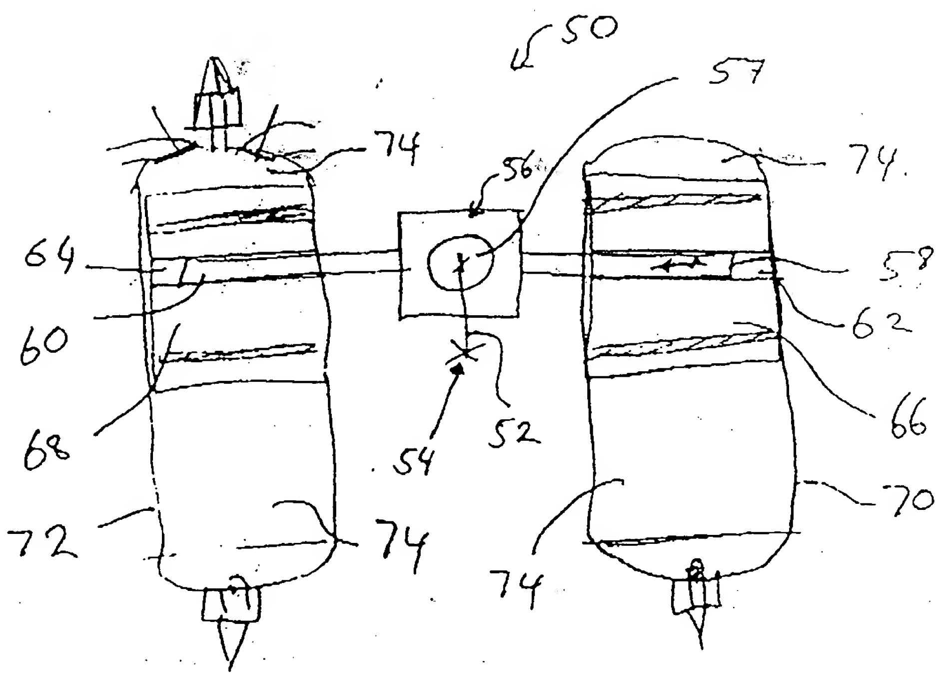


Fig 2



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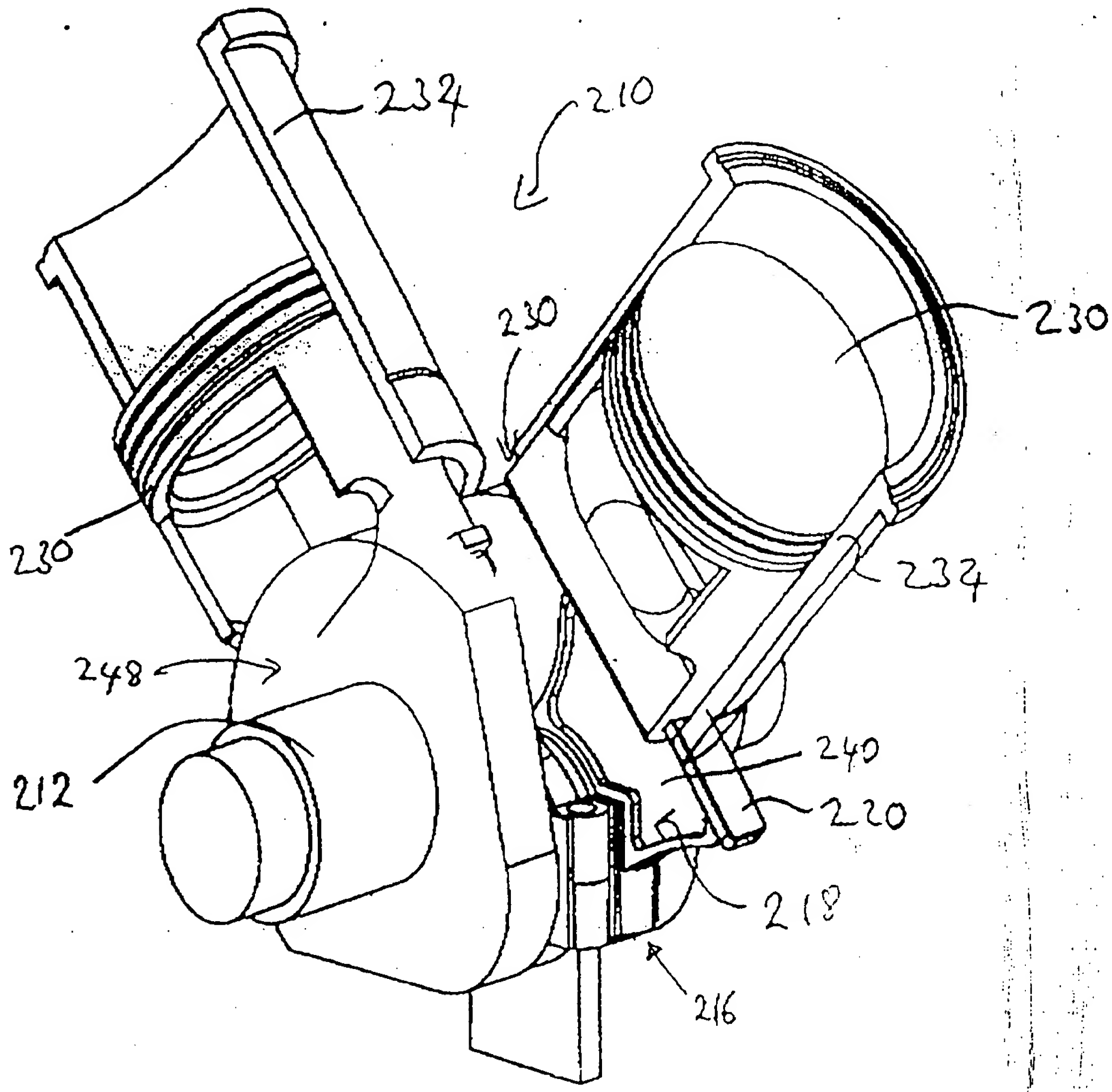

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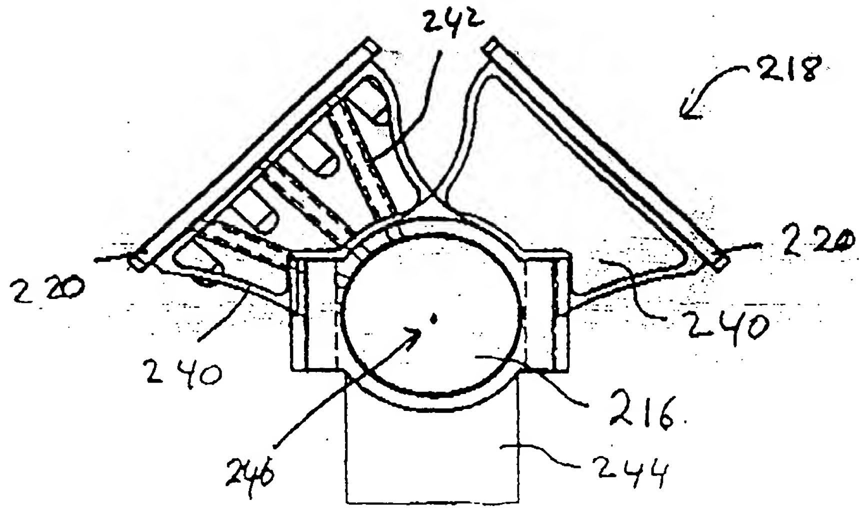
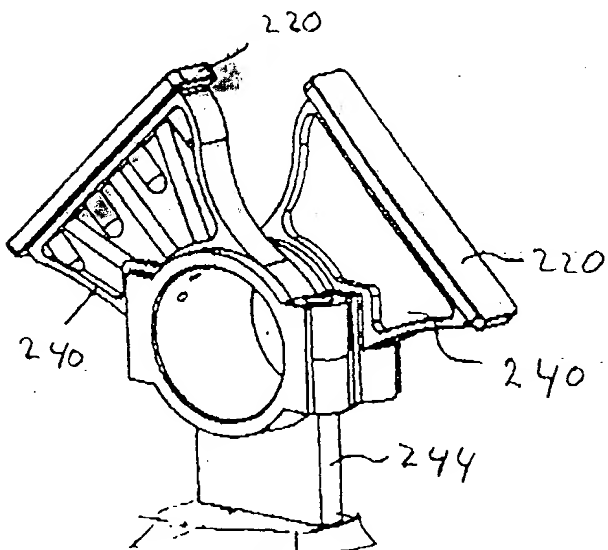


Fig 6



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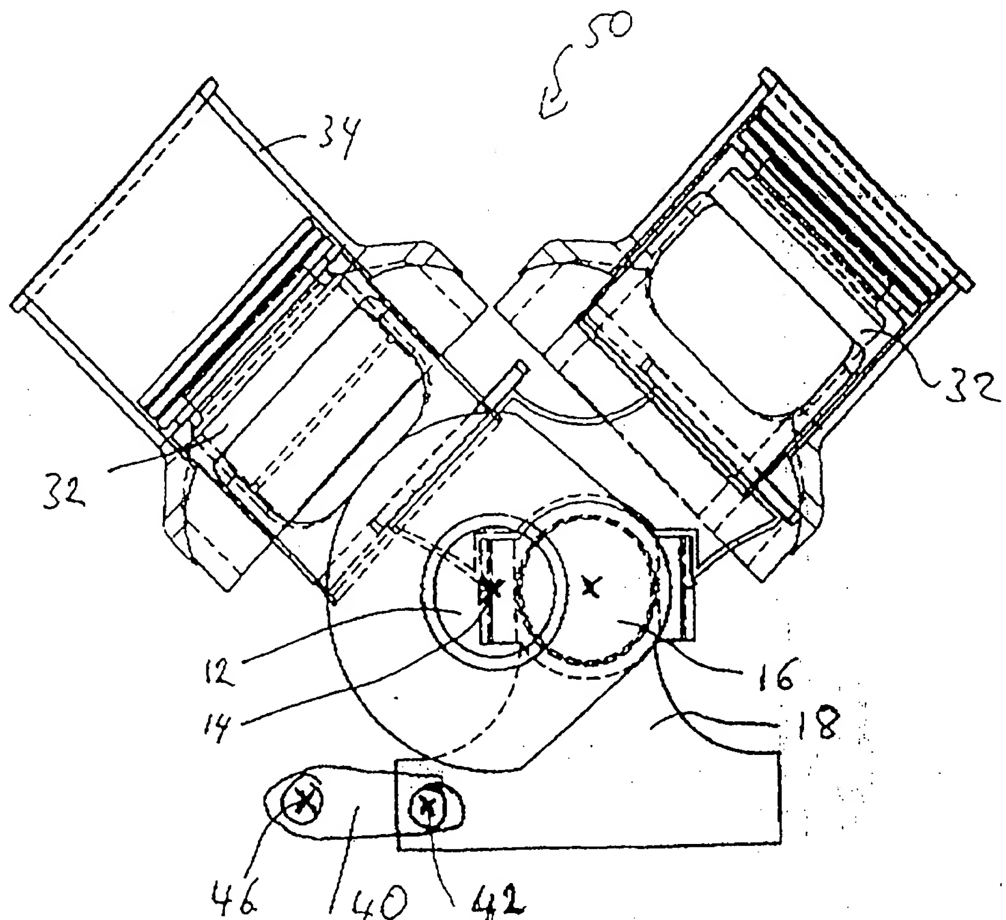

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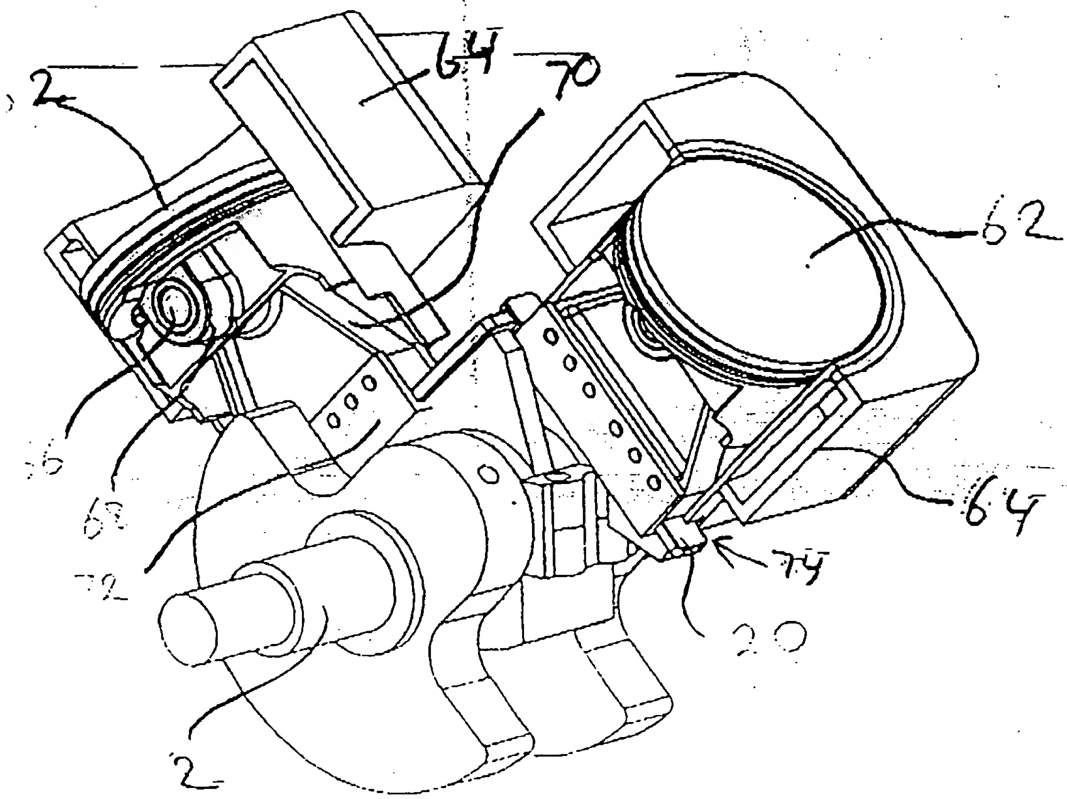

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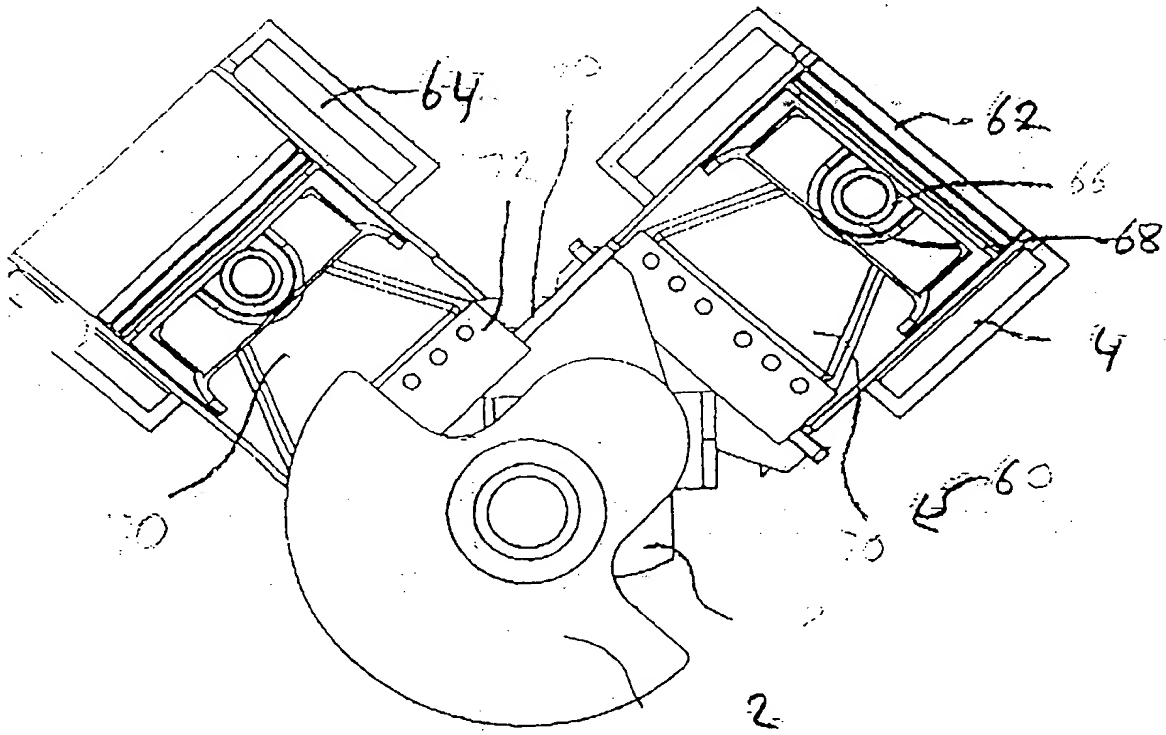


Fig 10

Fig 11

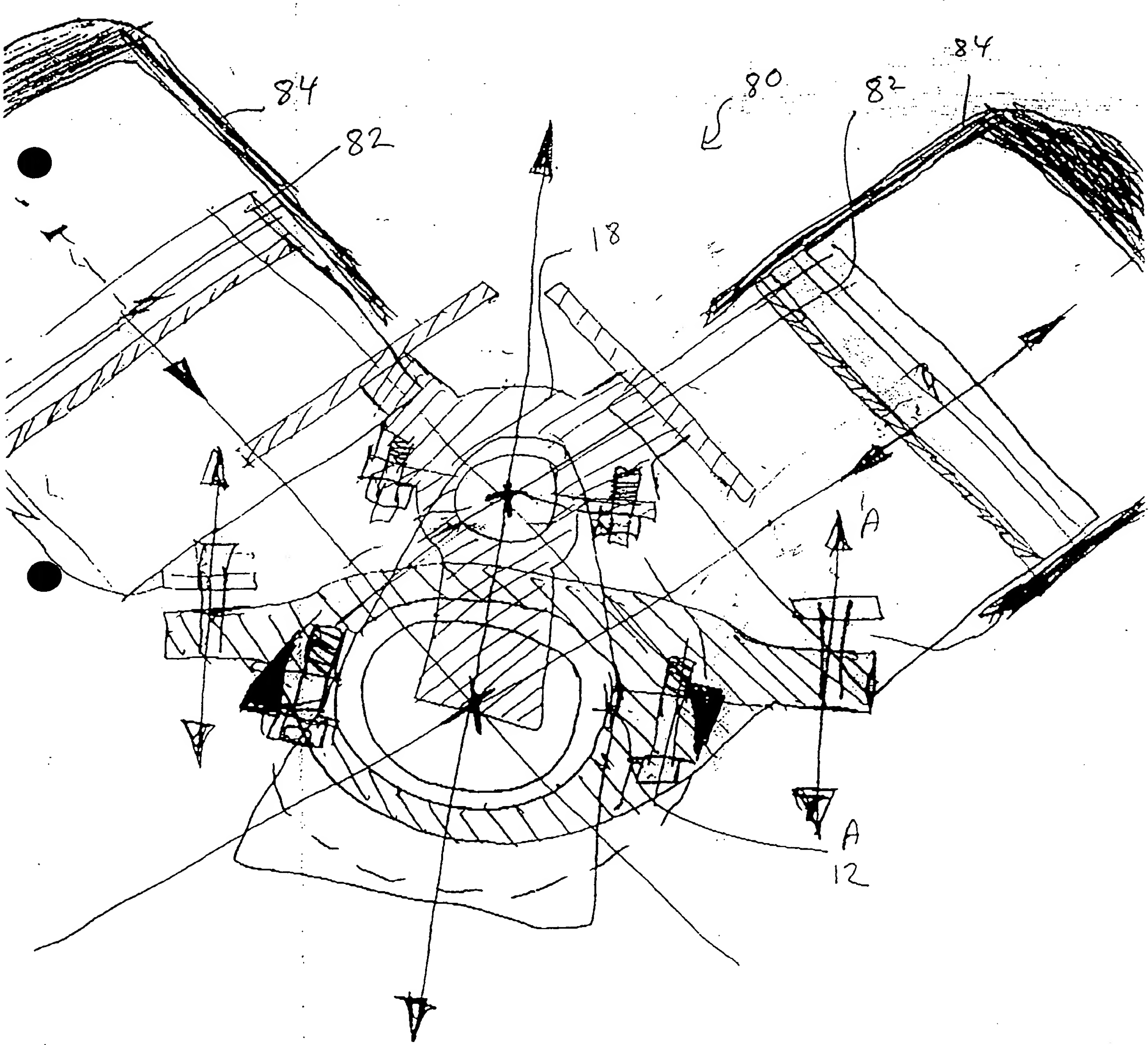
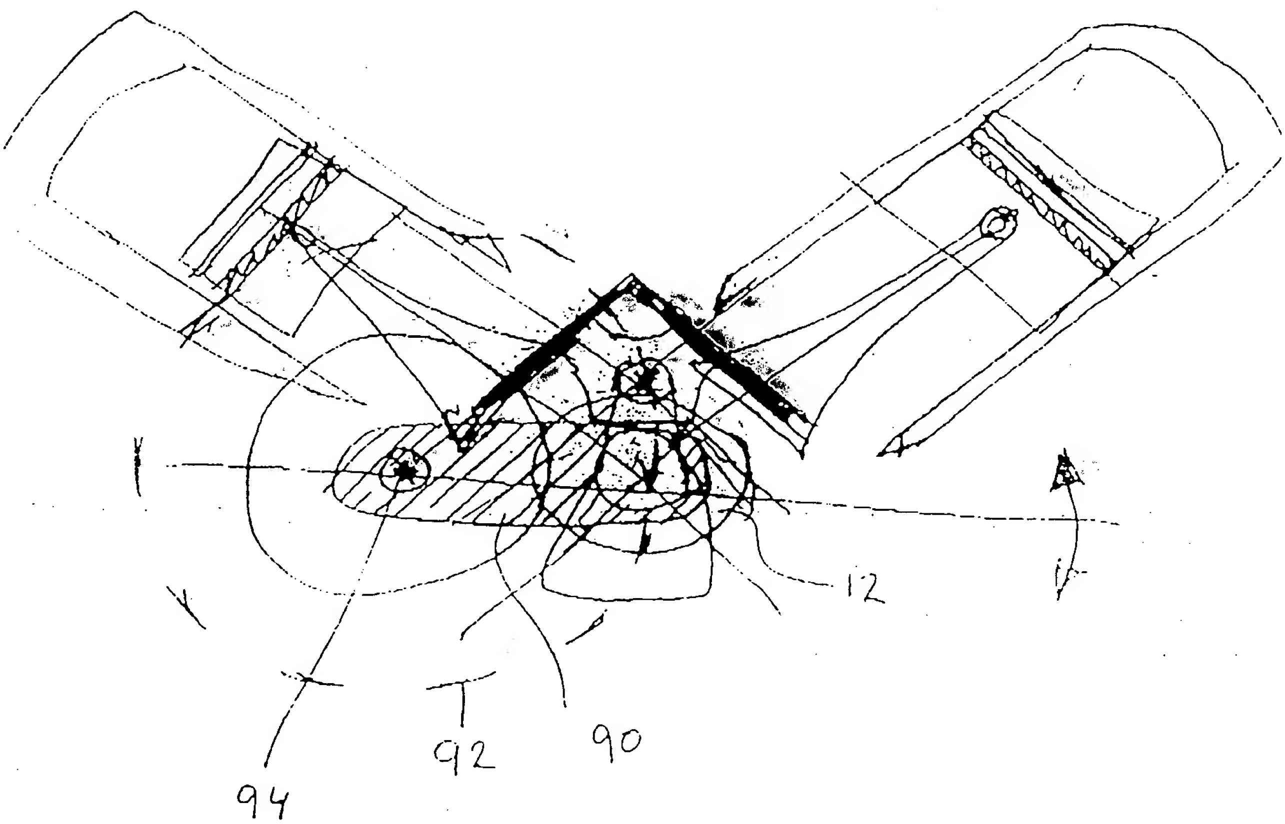


Fig 12



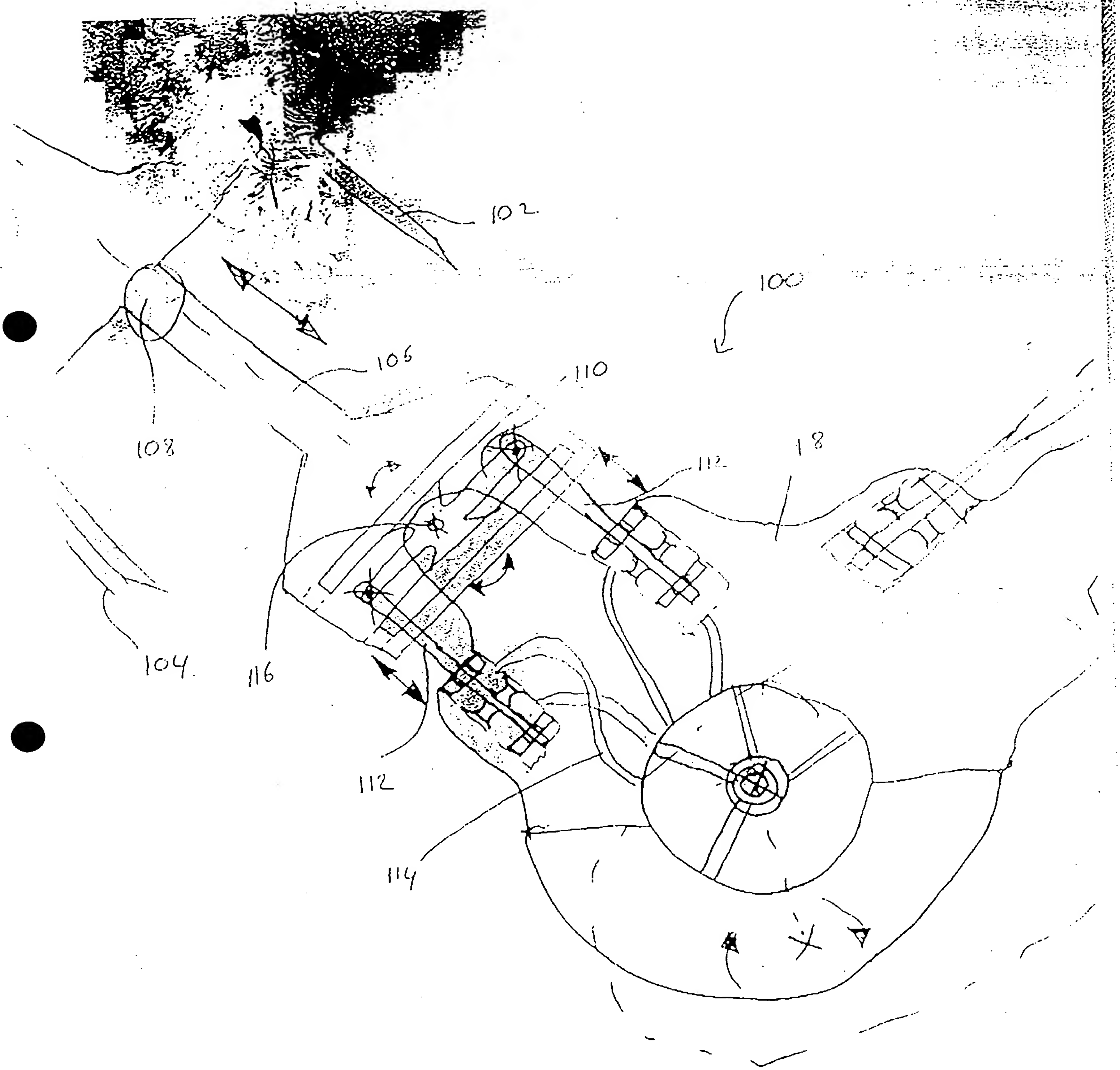


Fig 13

fig 21

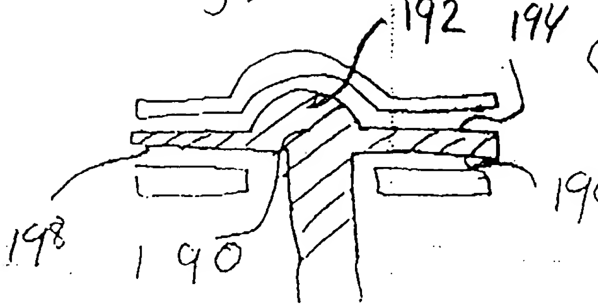
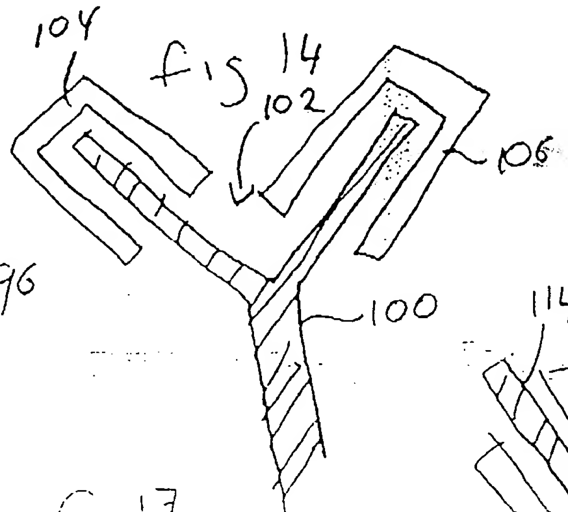


fig 14



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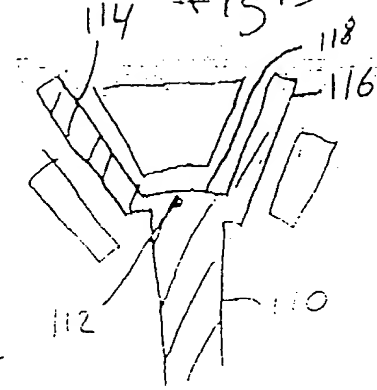

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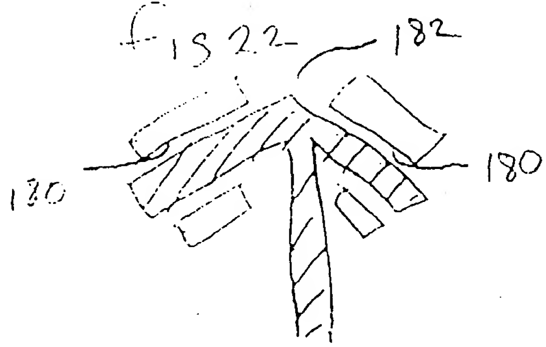
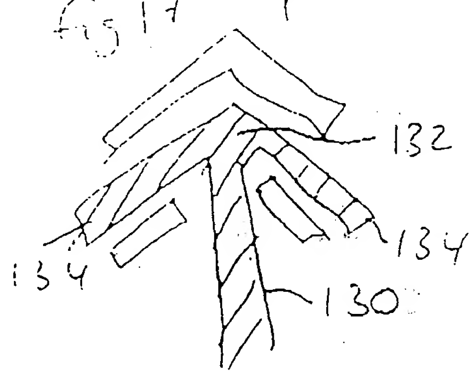


fig 17



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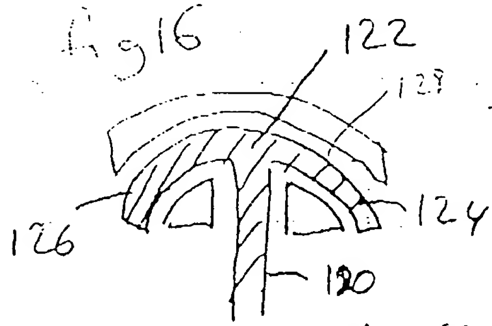
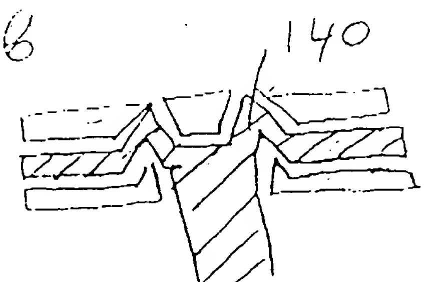


fig 16



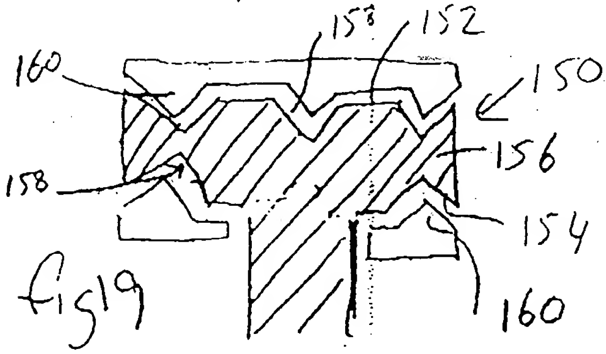


fig19

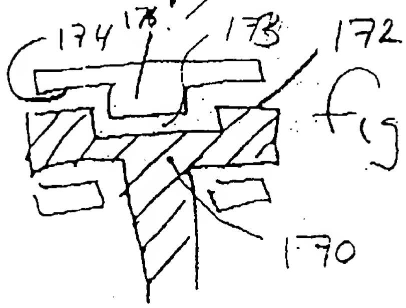
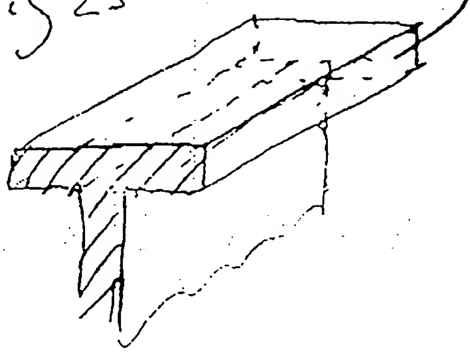


fig 20

fig 23



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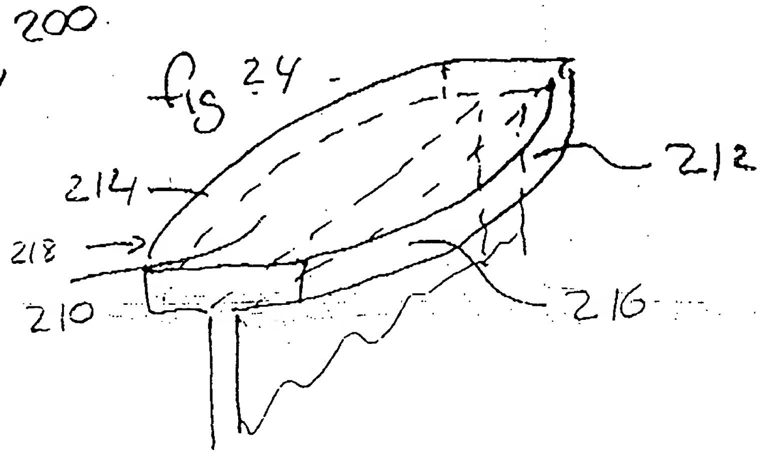

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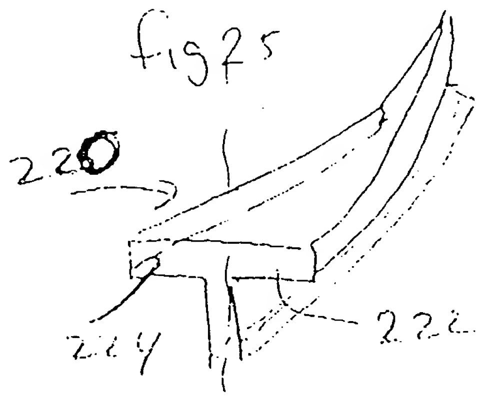

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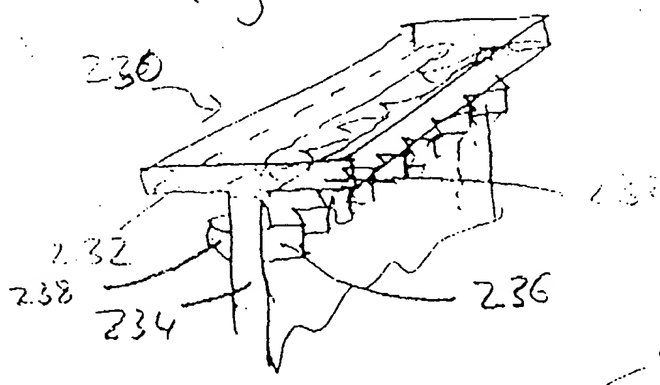


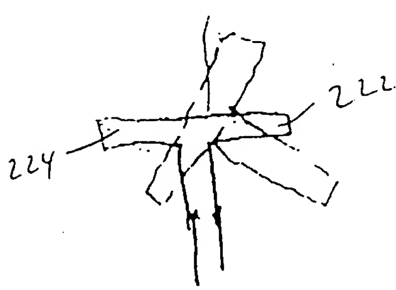
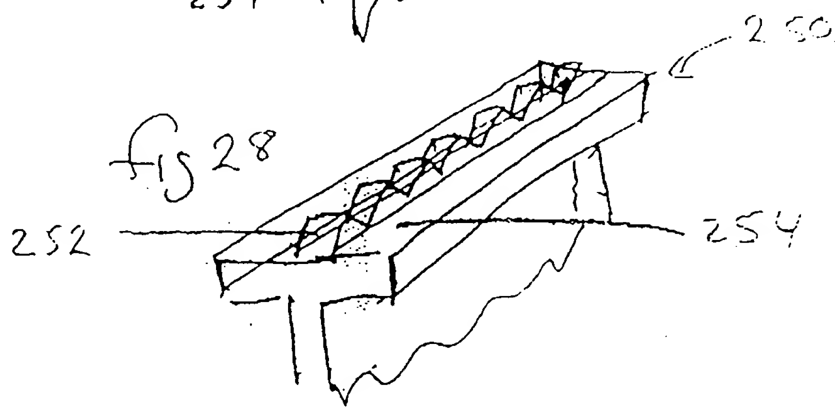


fig 27

fig 28



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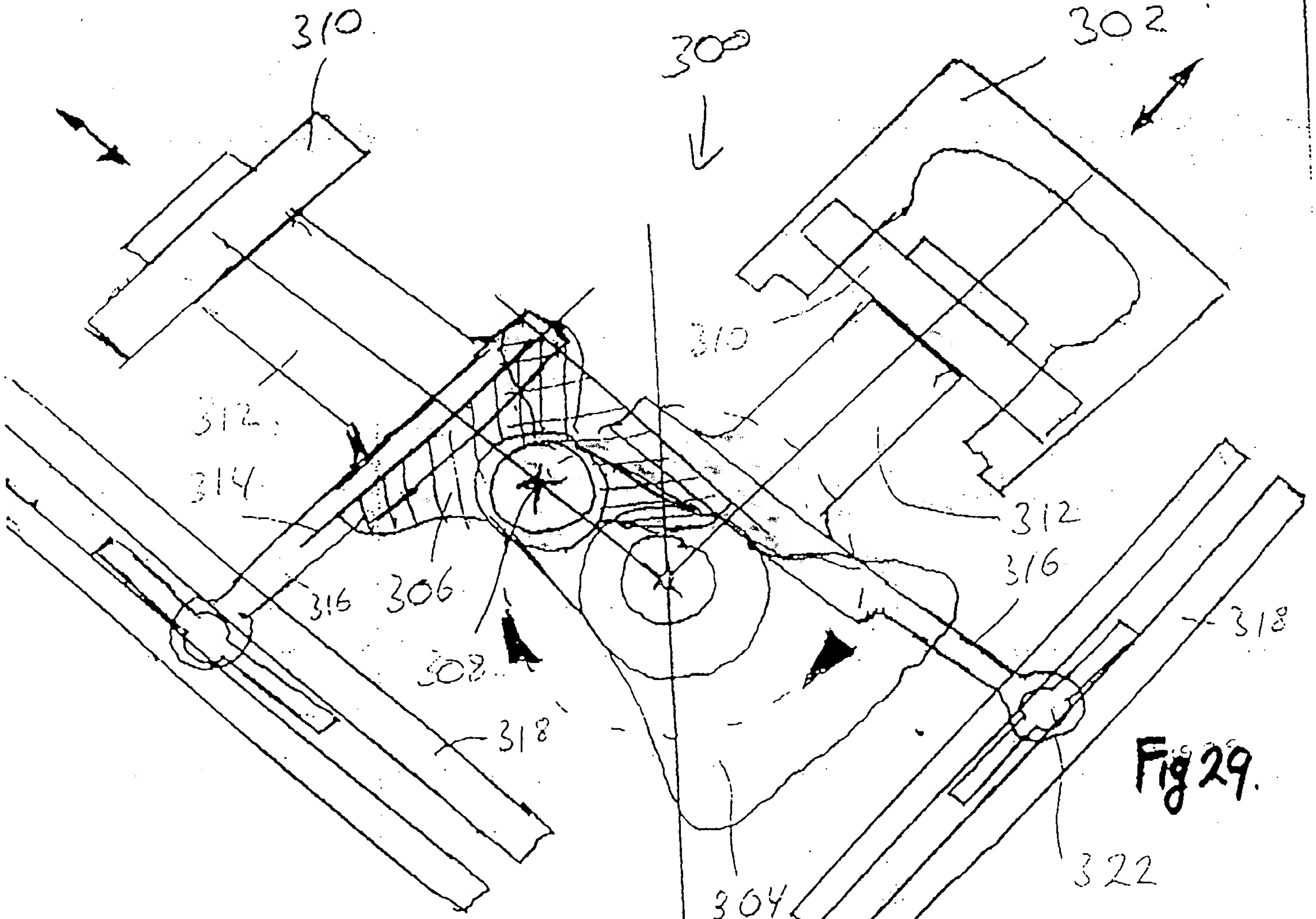

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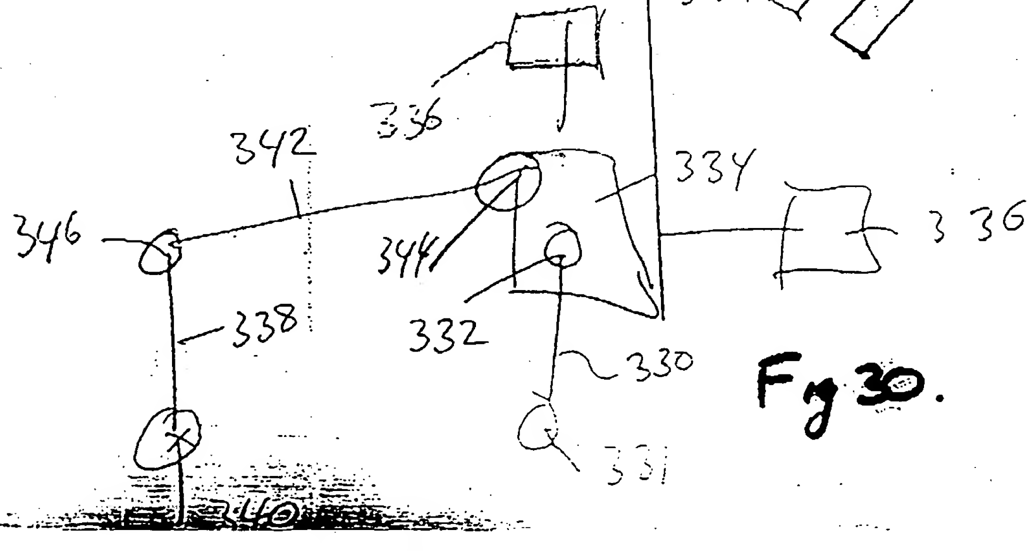
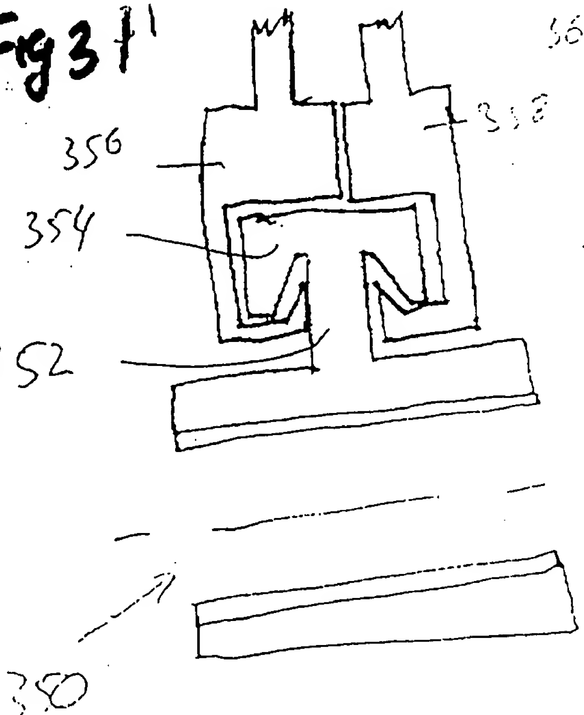


Fig 30.

Fig 31



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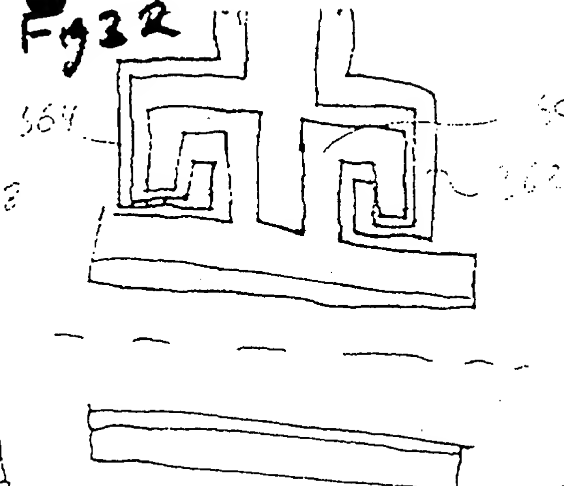

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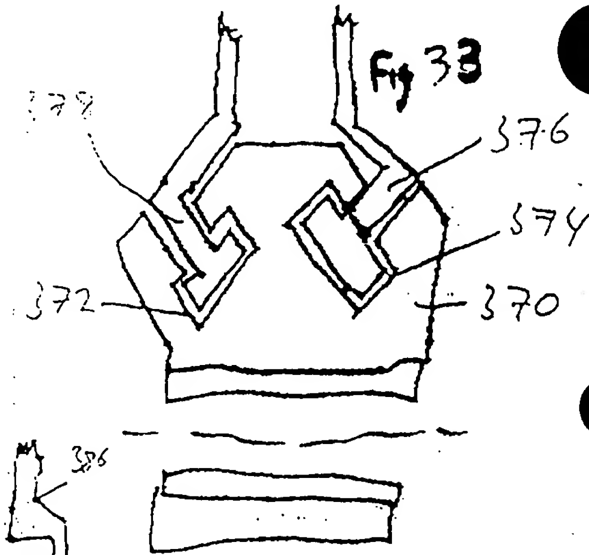

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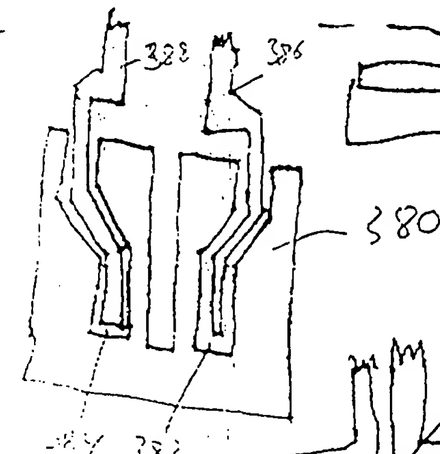

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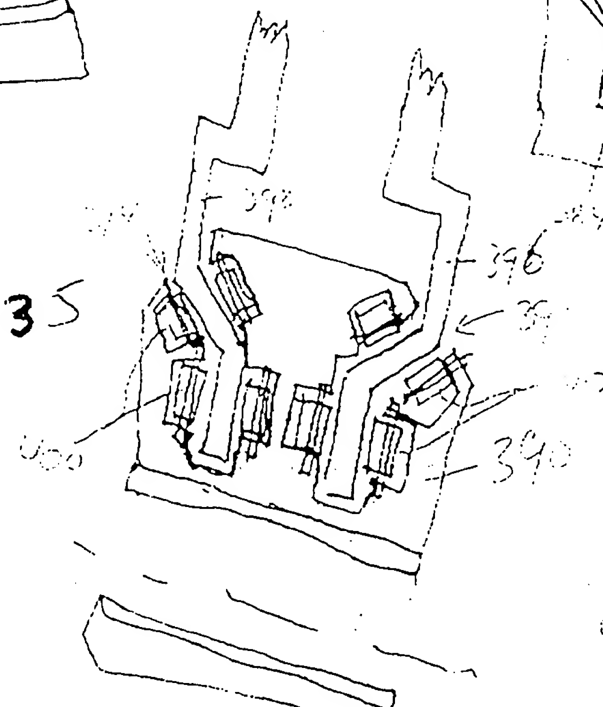

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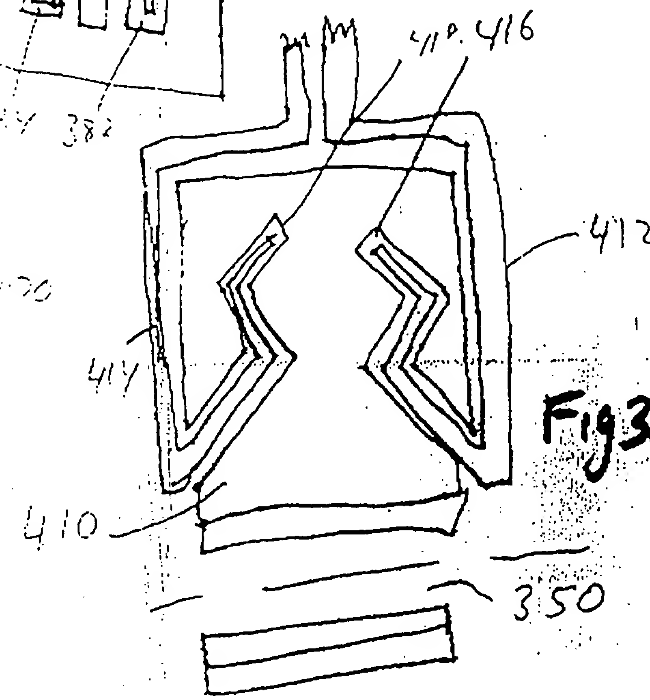

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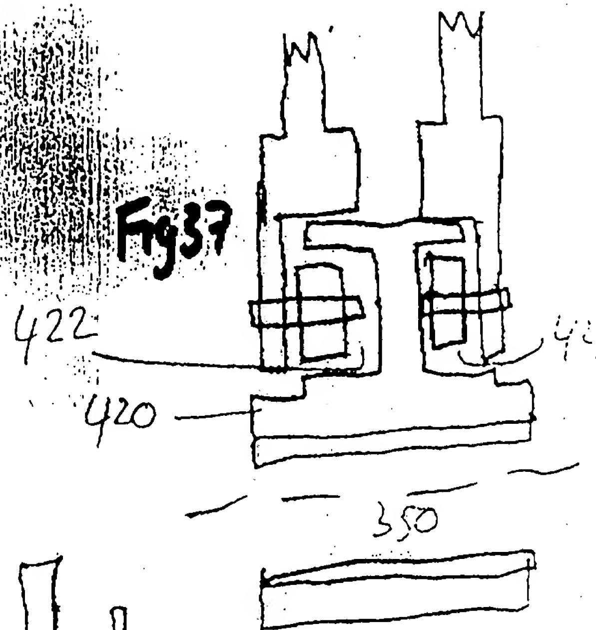

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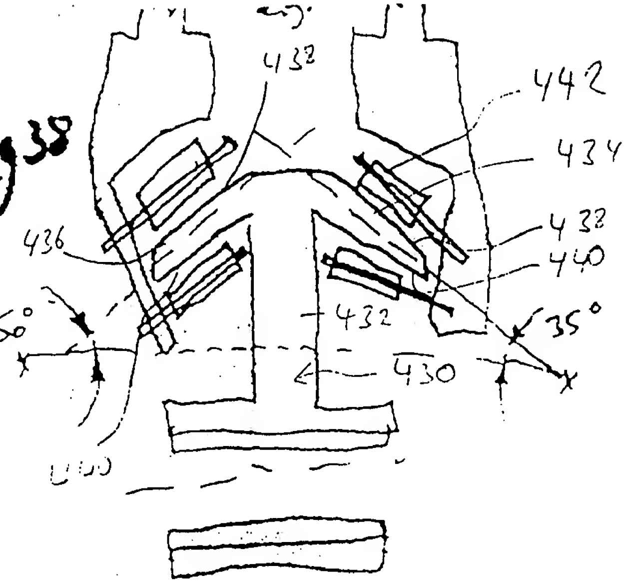

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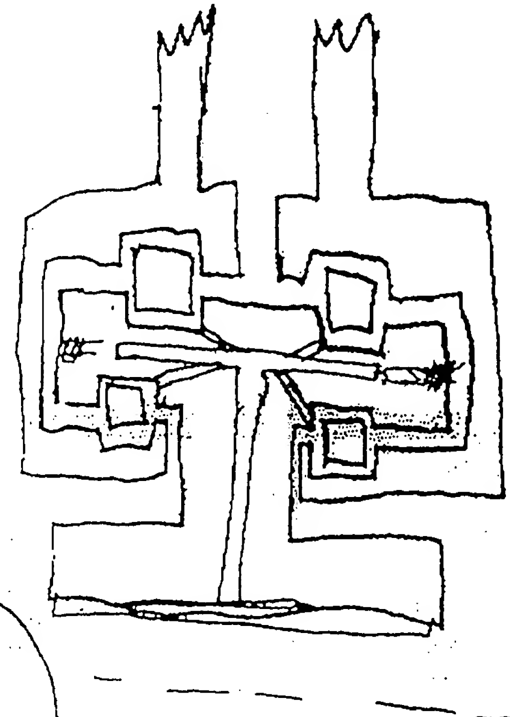

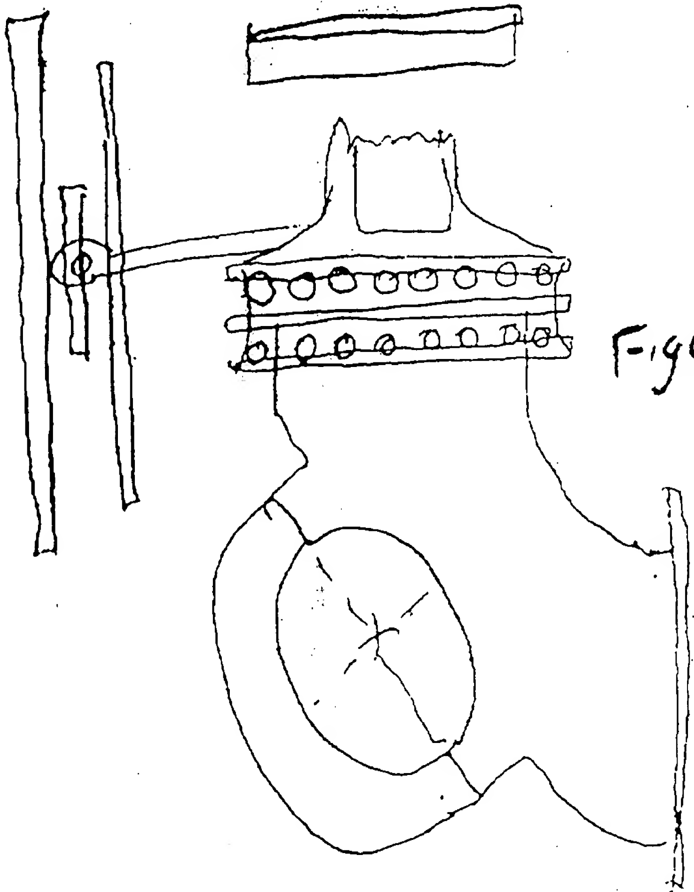

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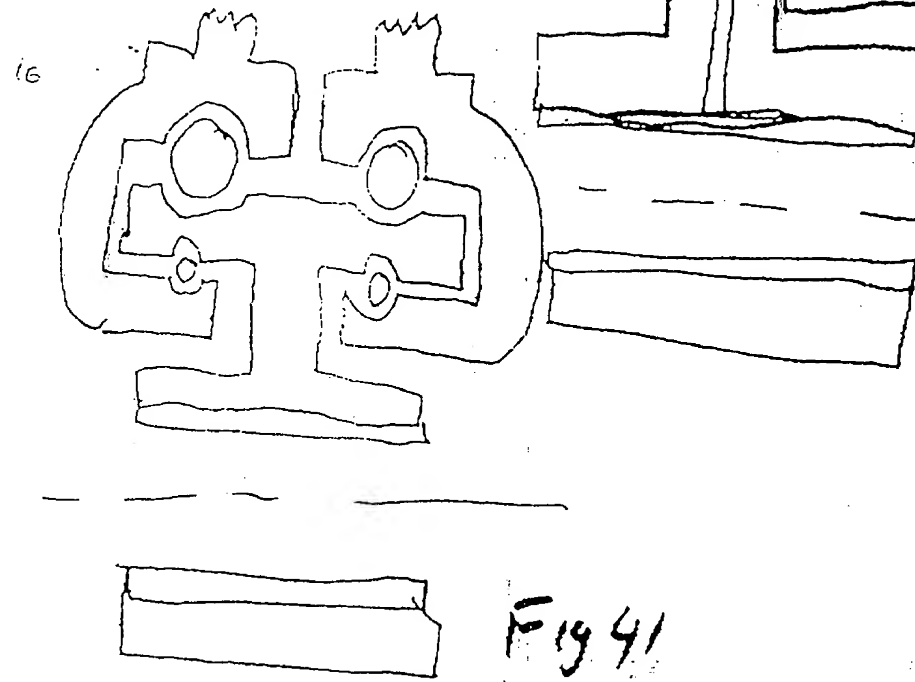
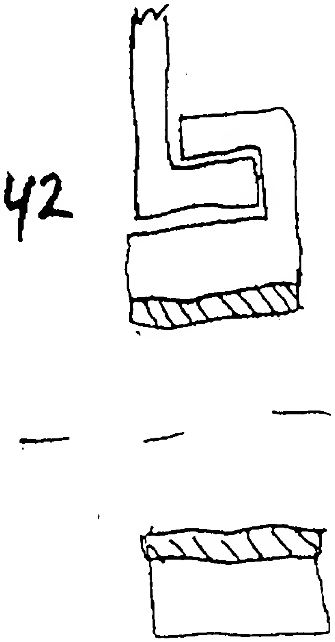


Fig 41

Fig 42



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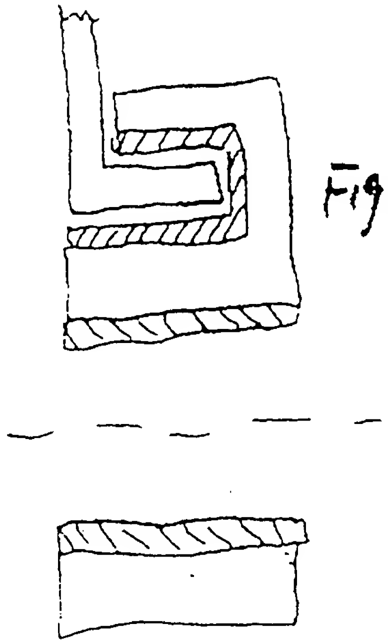

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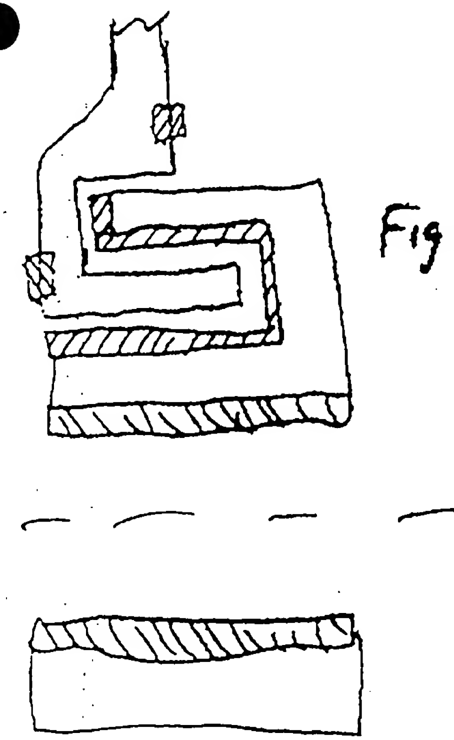

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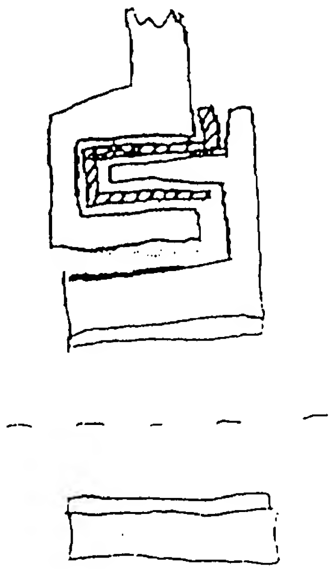

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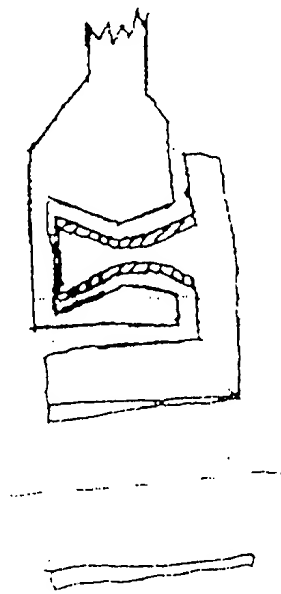
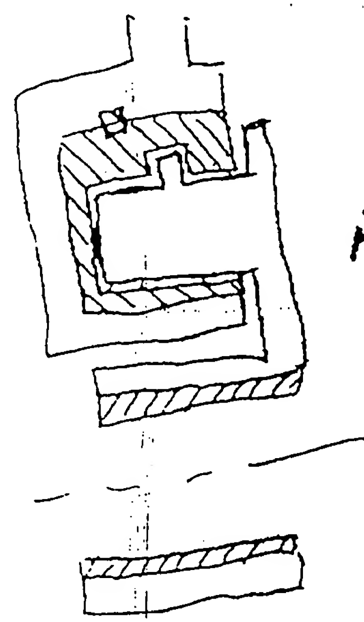


Fig 47



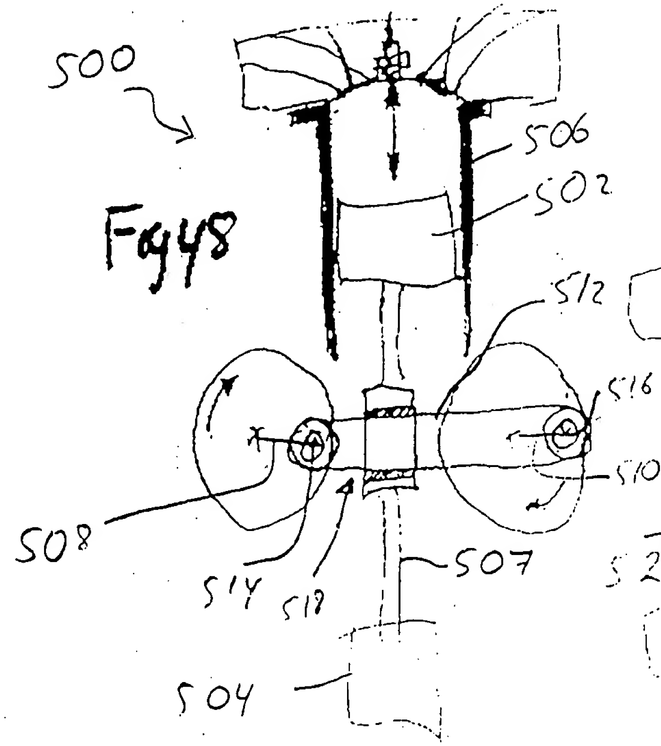


Fig 48

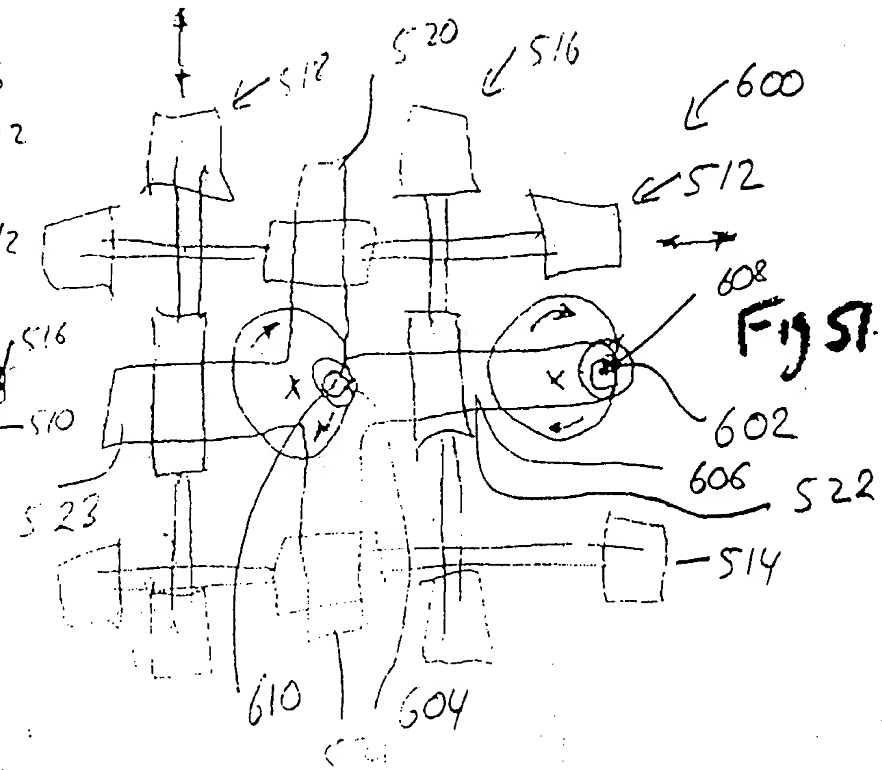
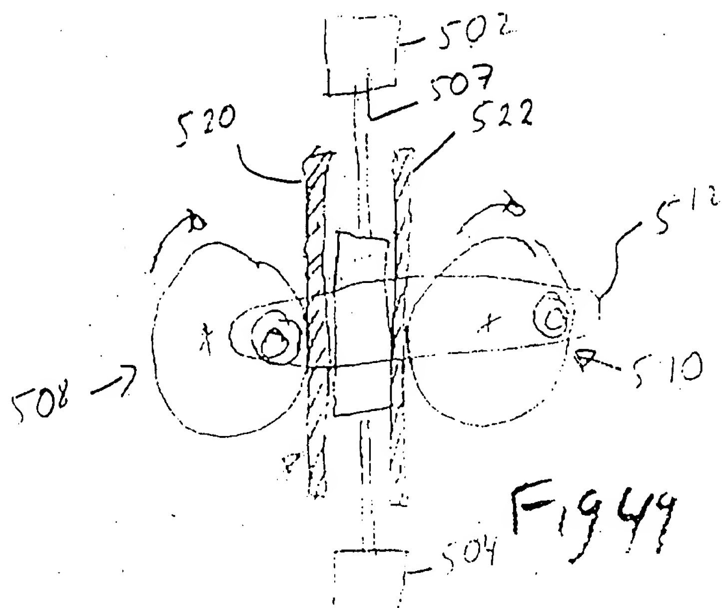


Fig 51



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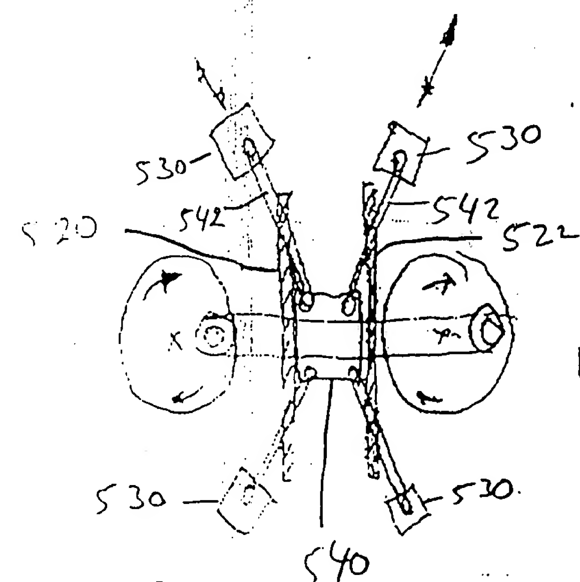

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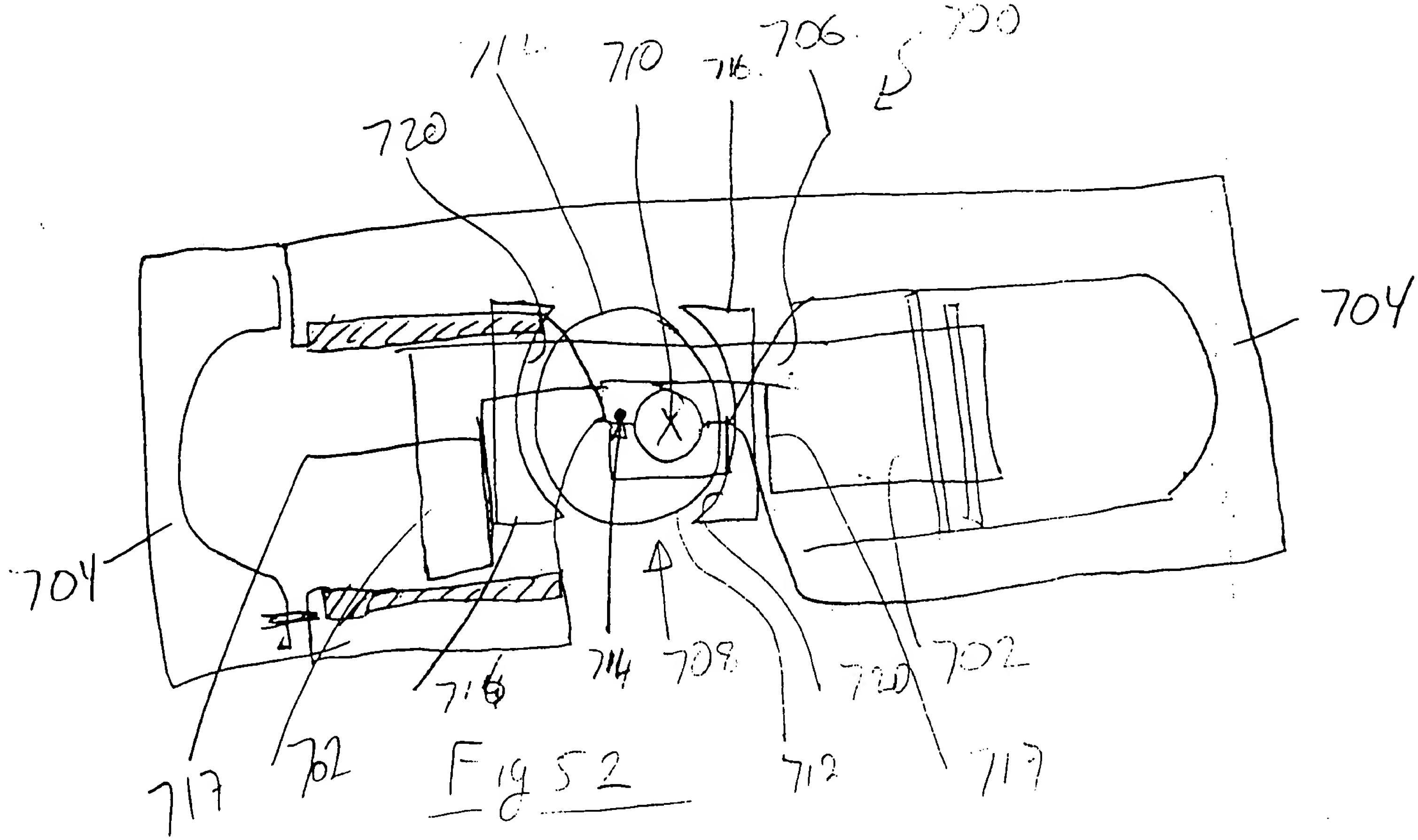

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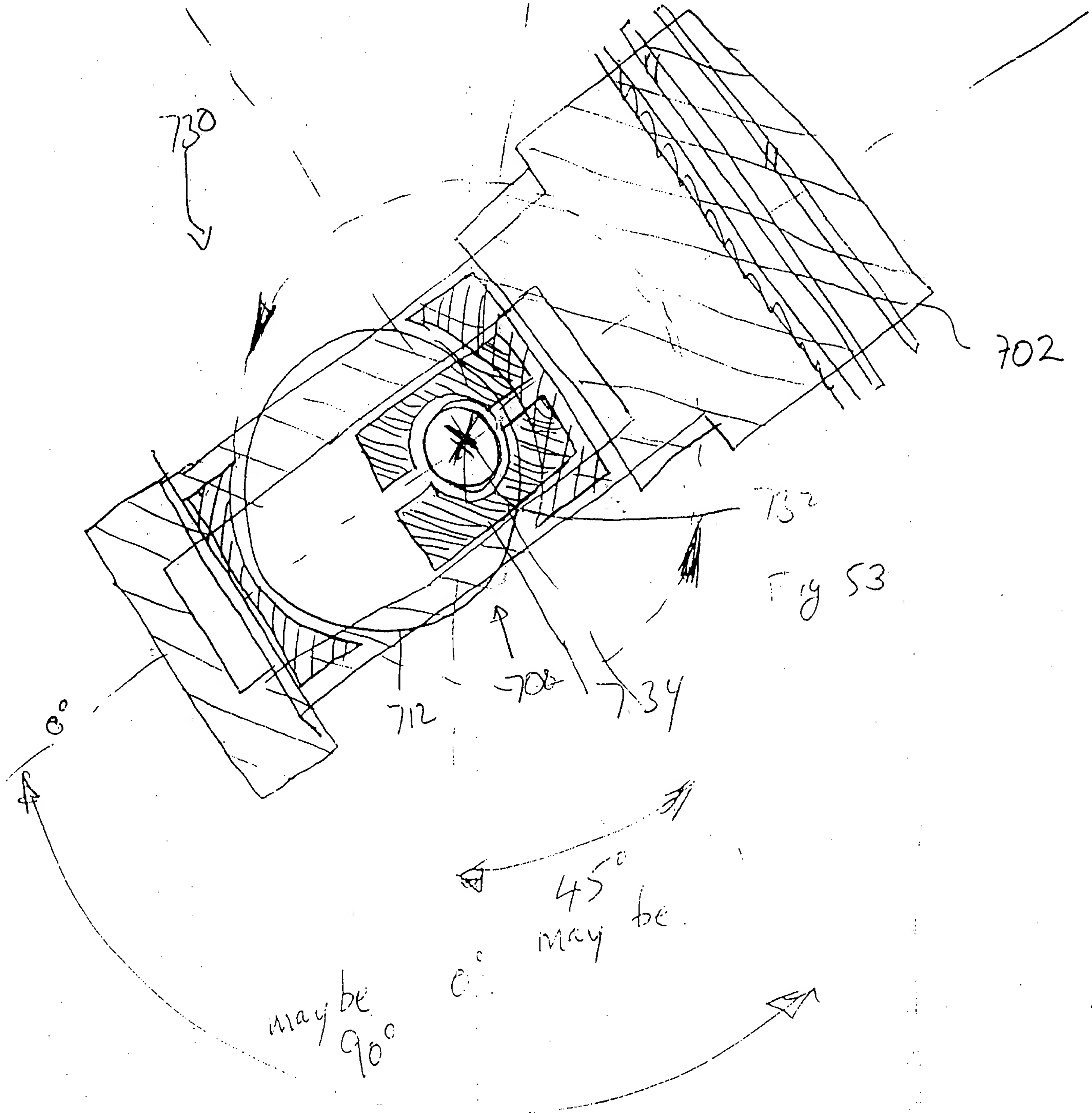

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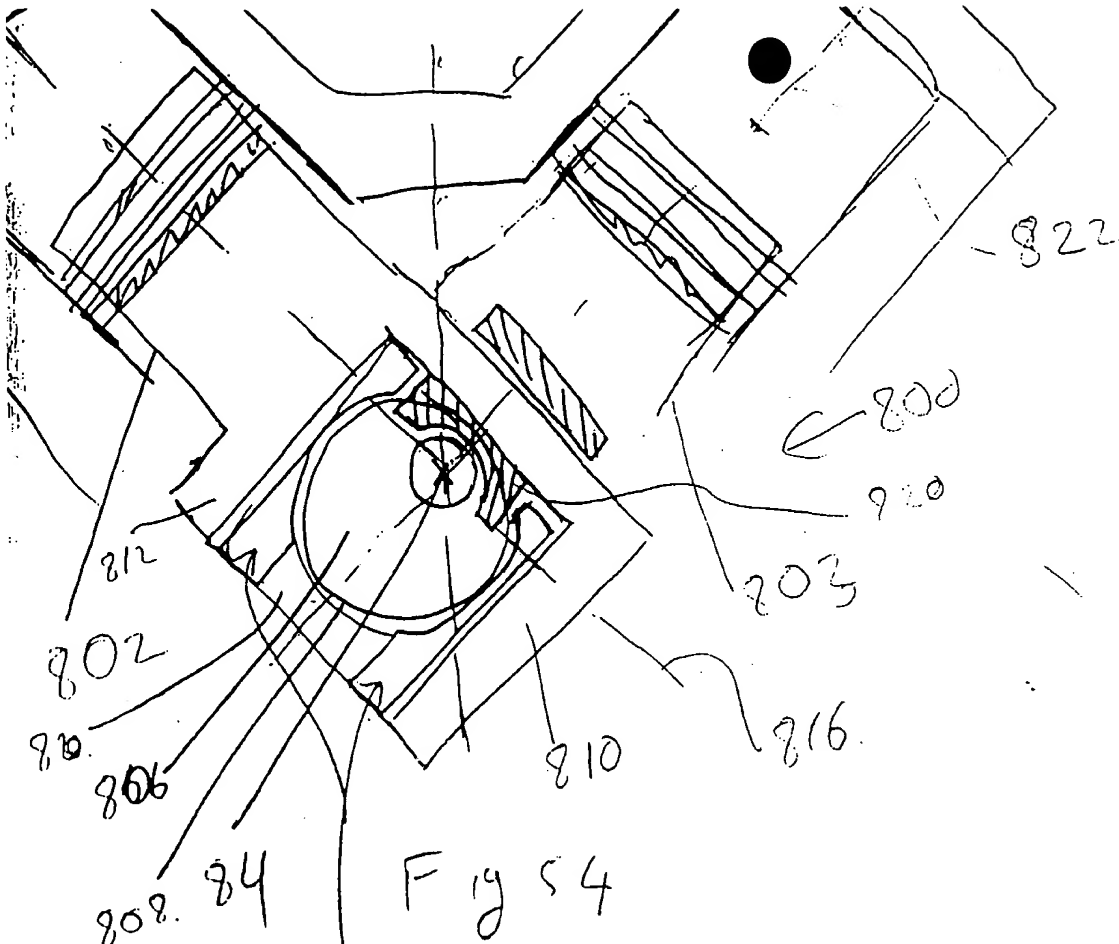

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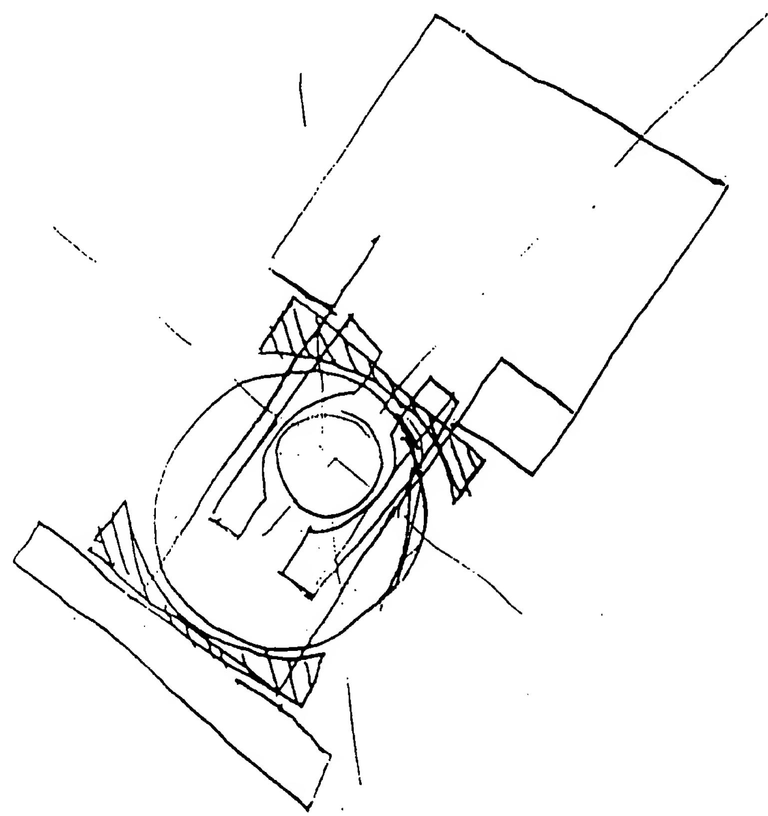

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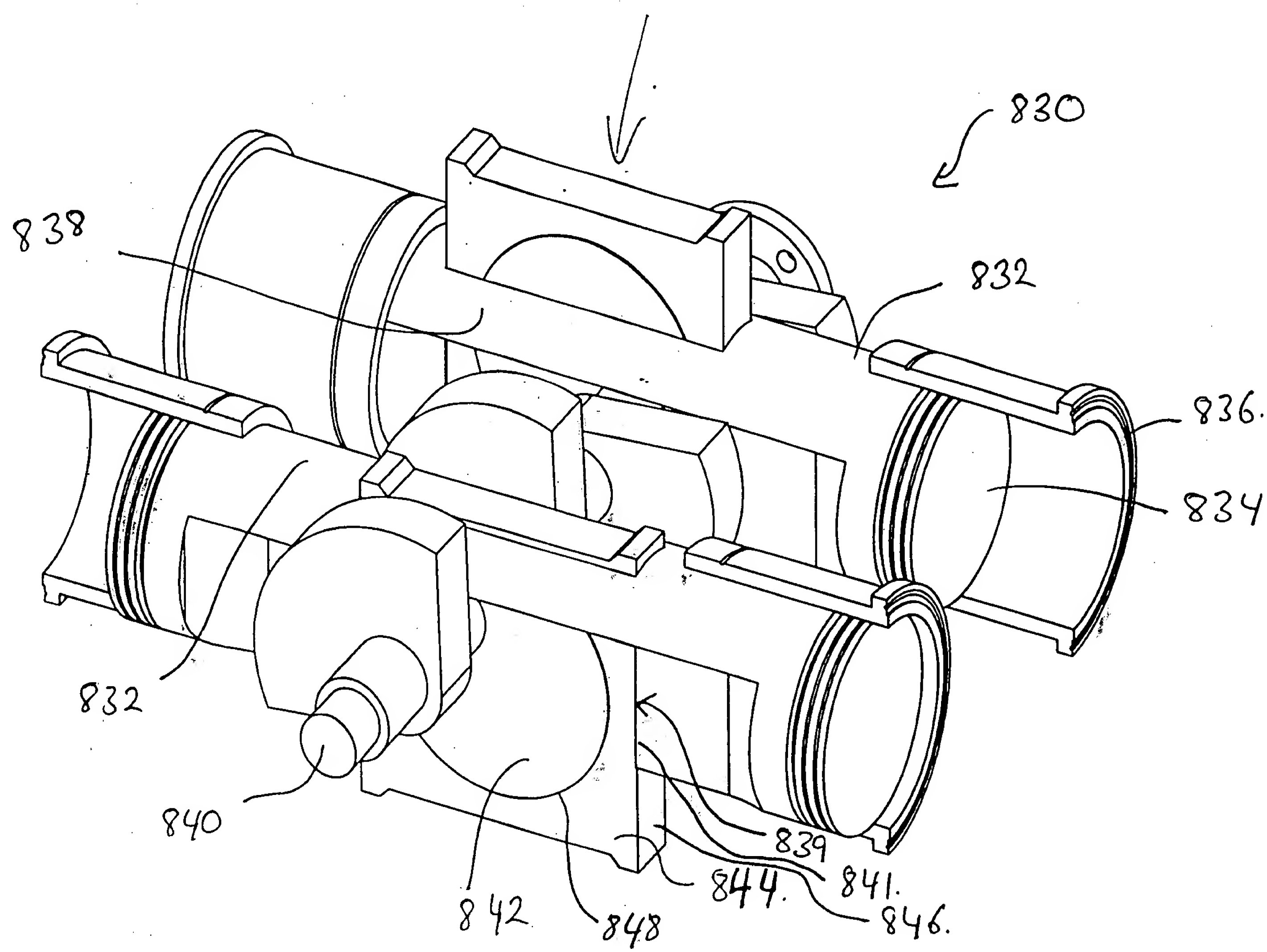

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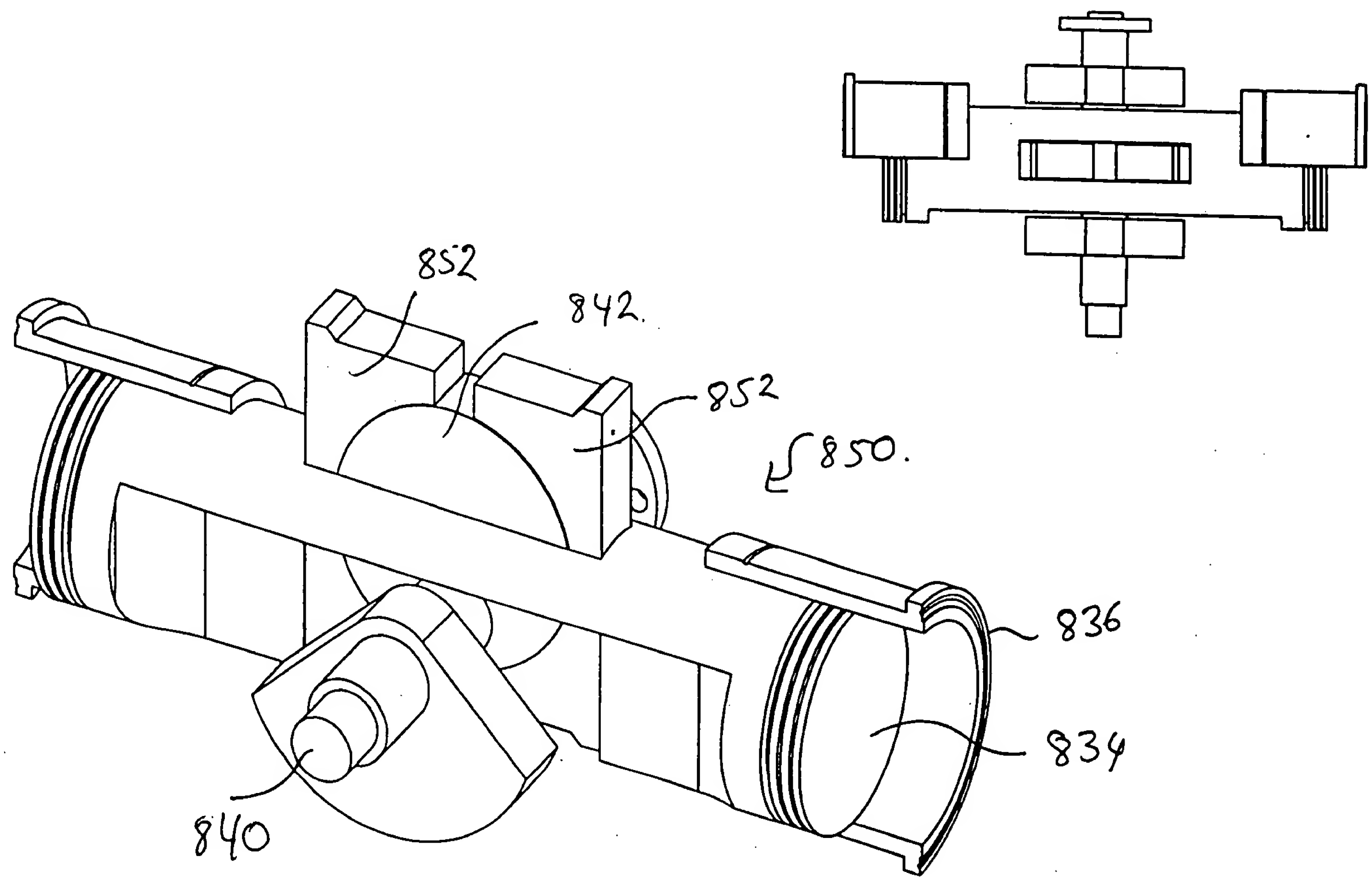

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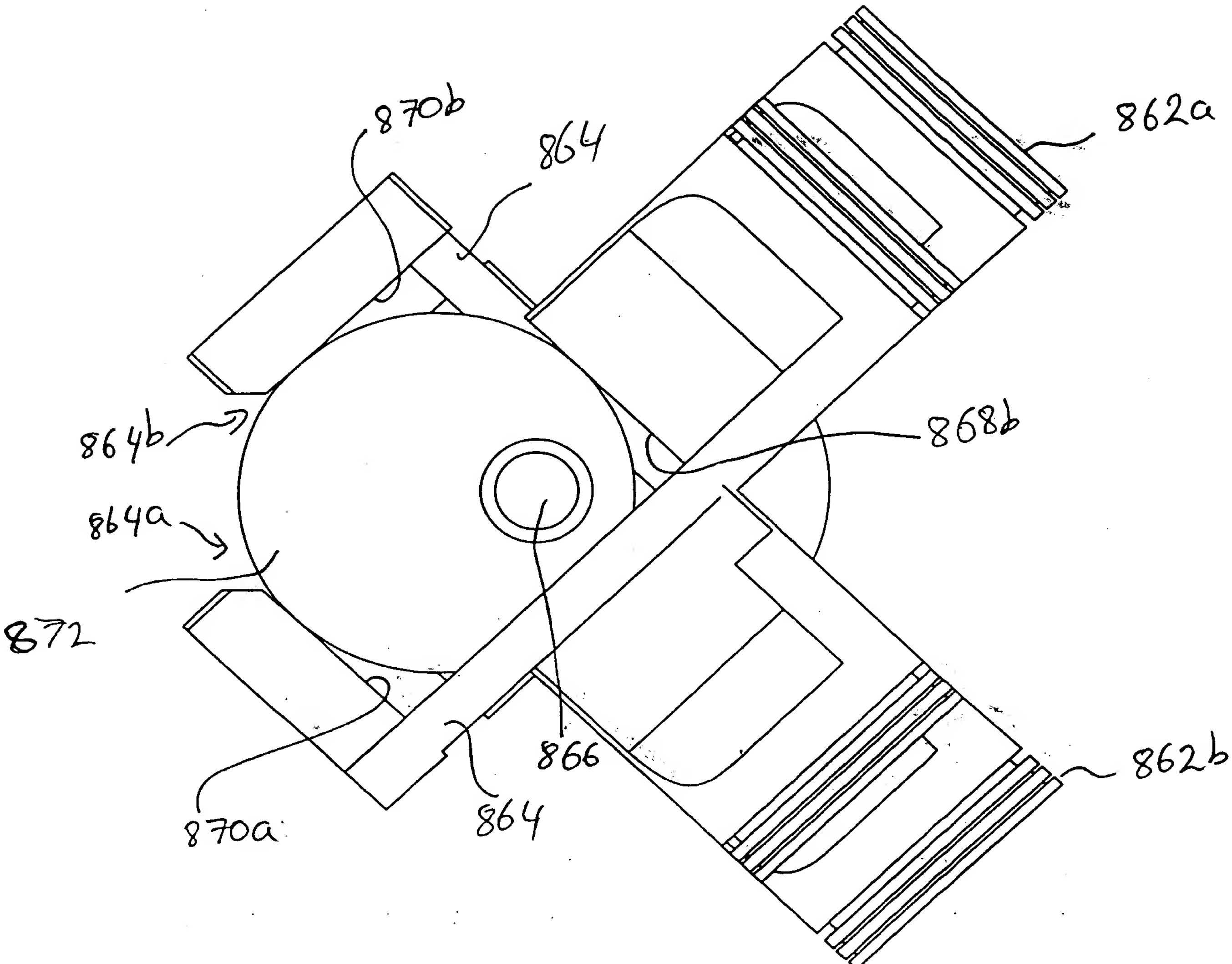

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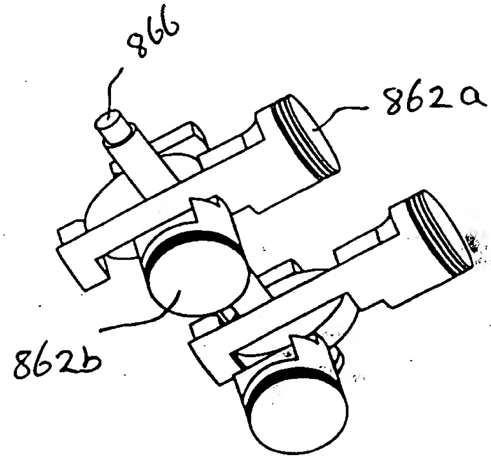

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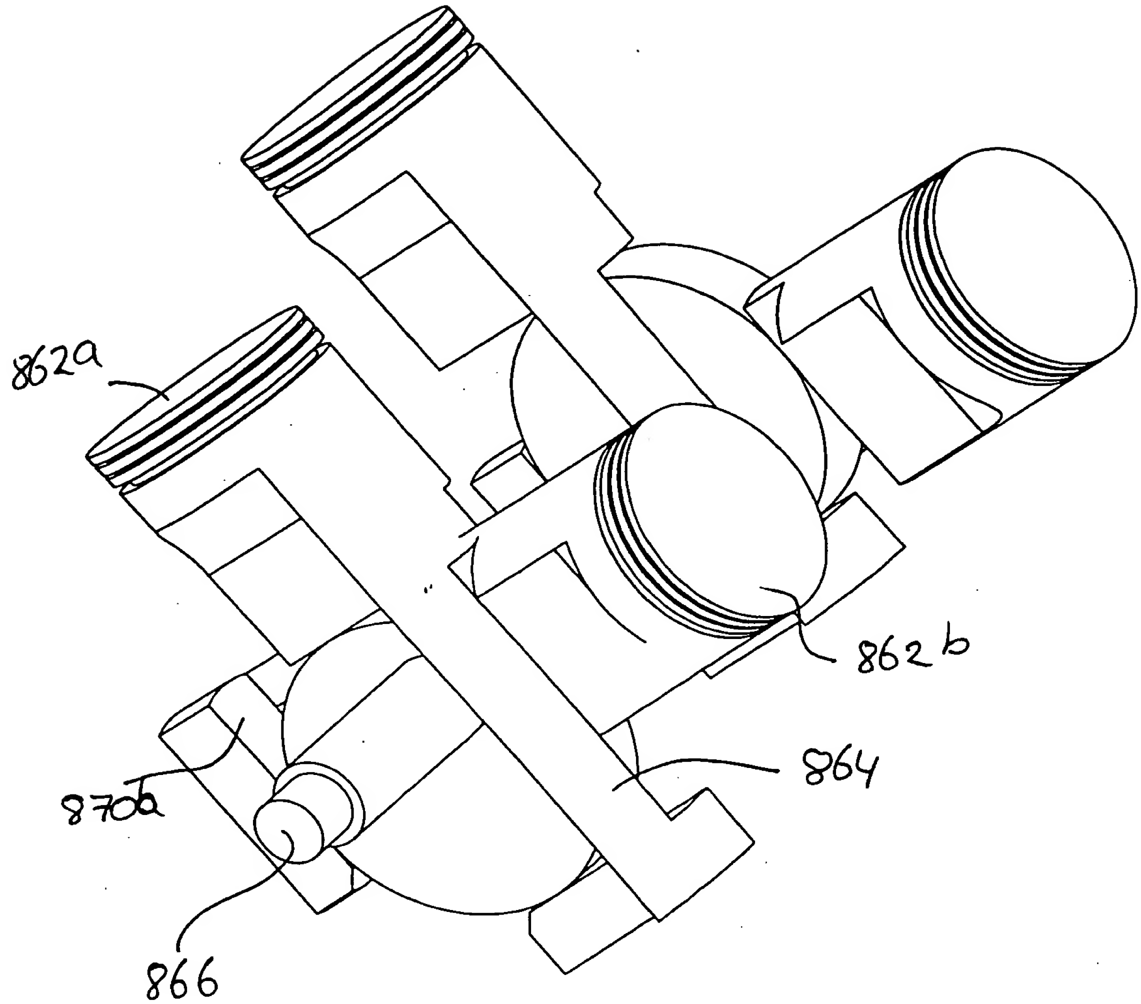

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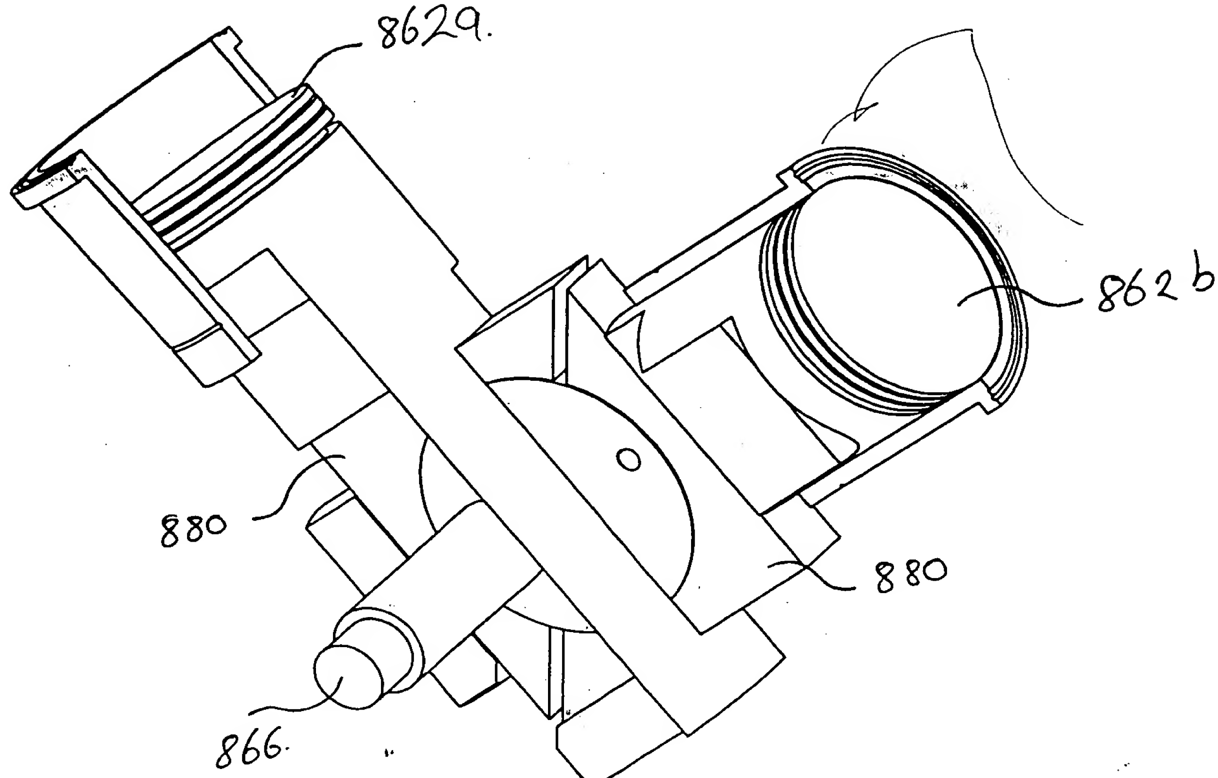

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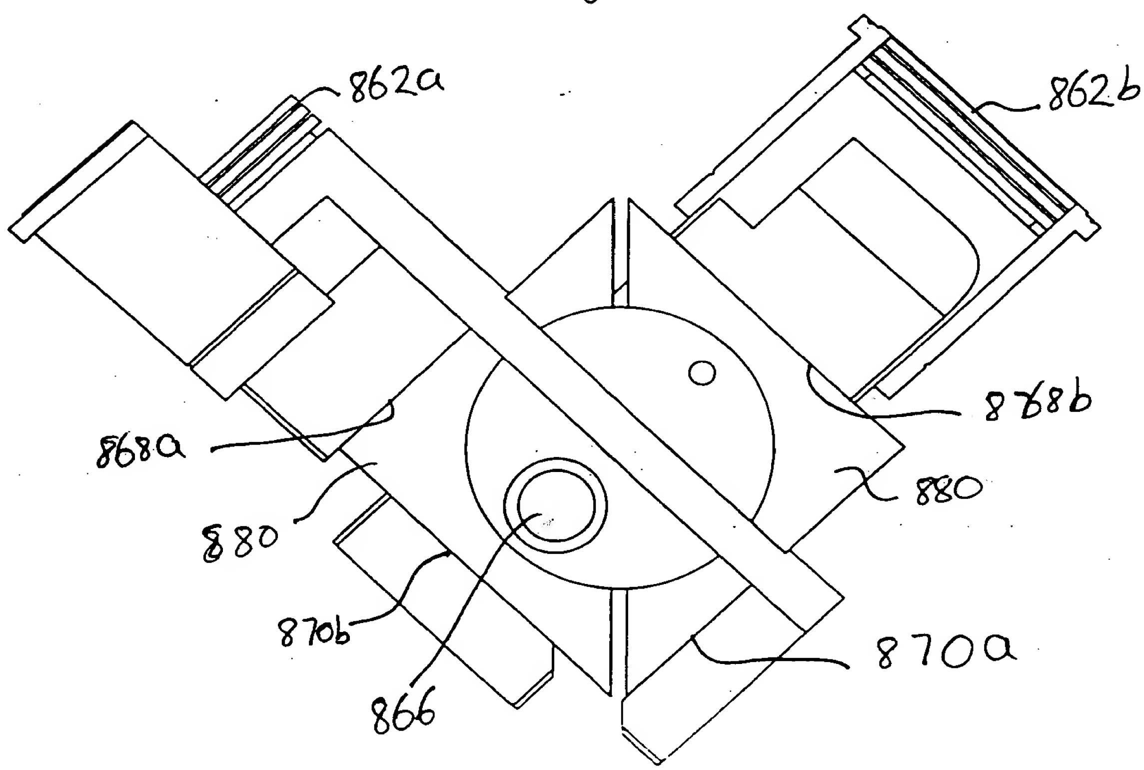

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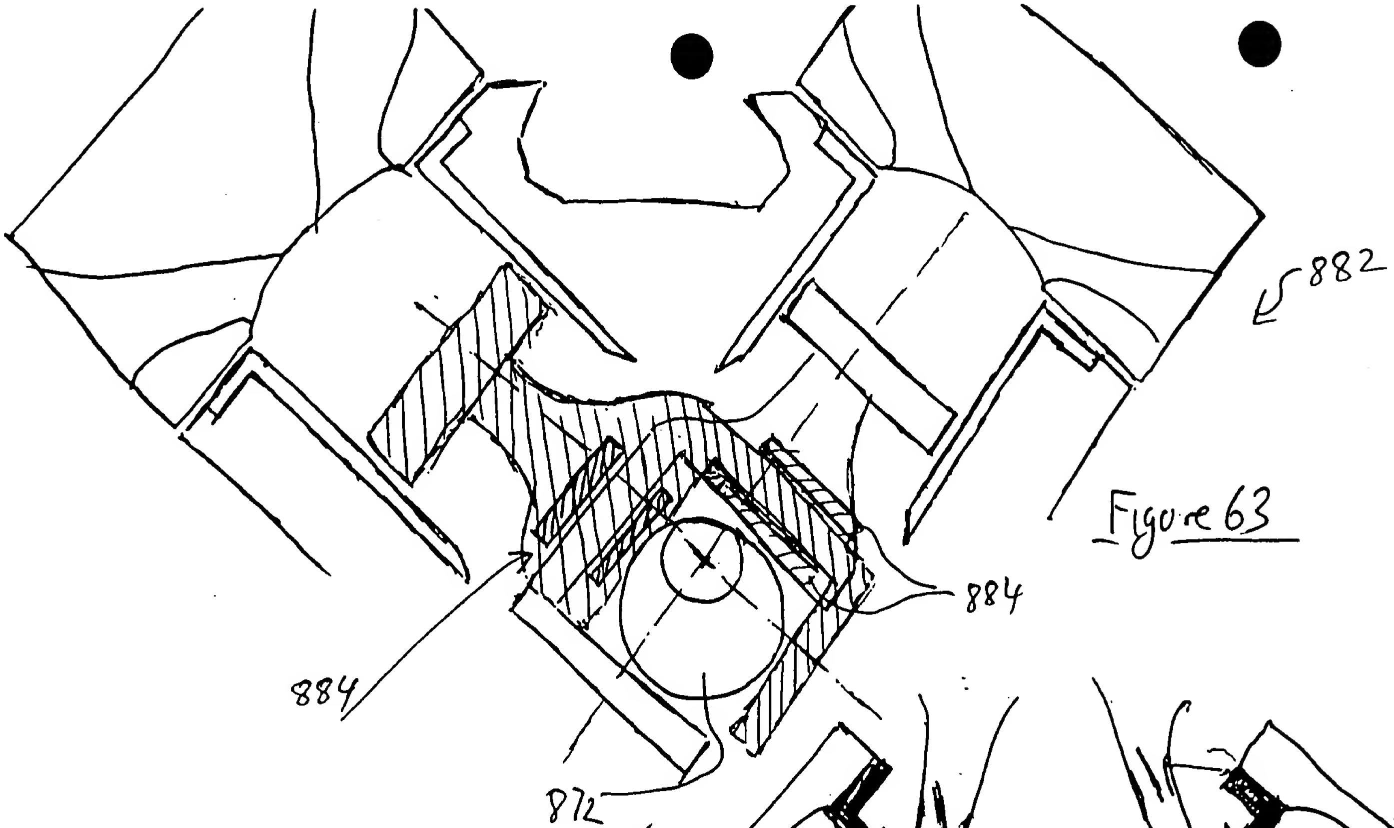

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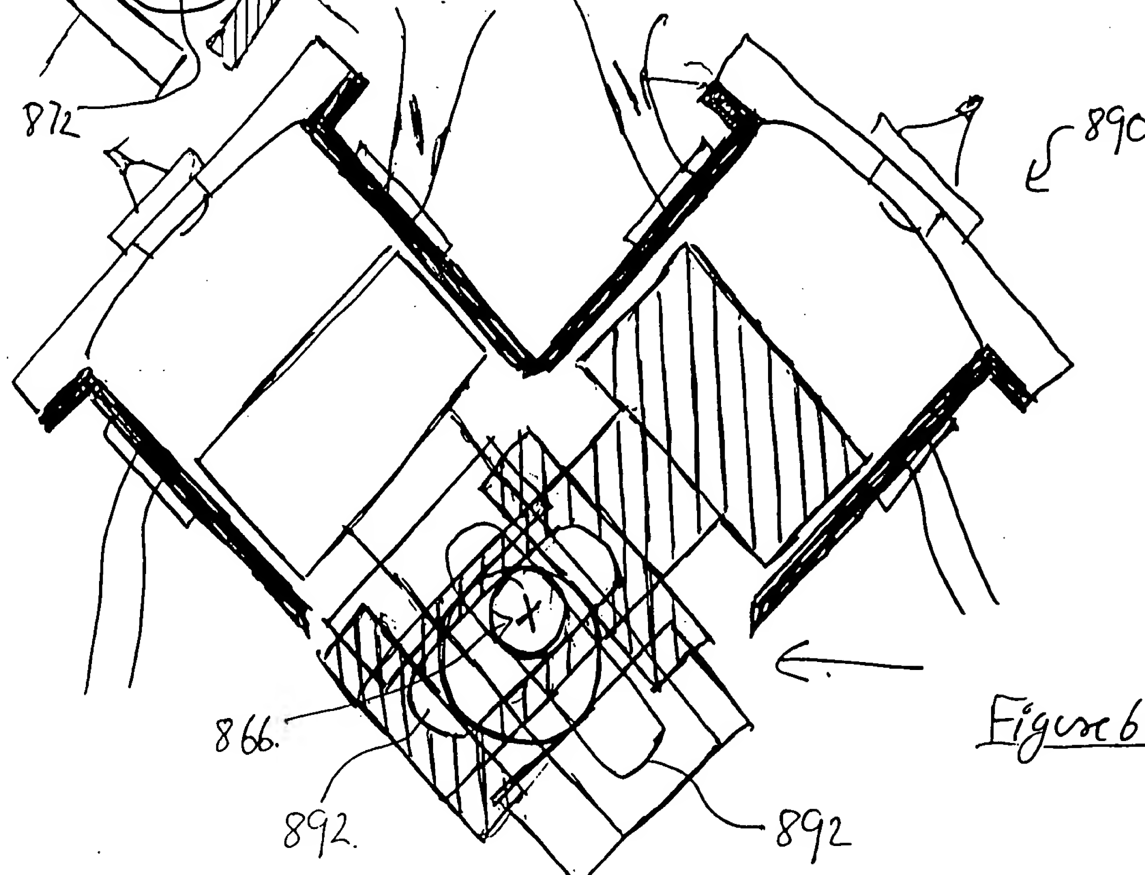

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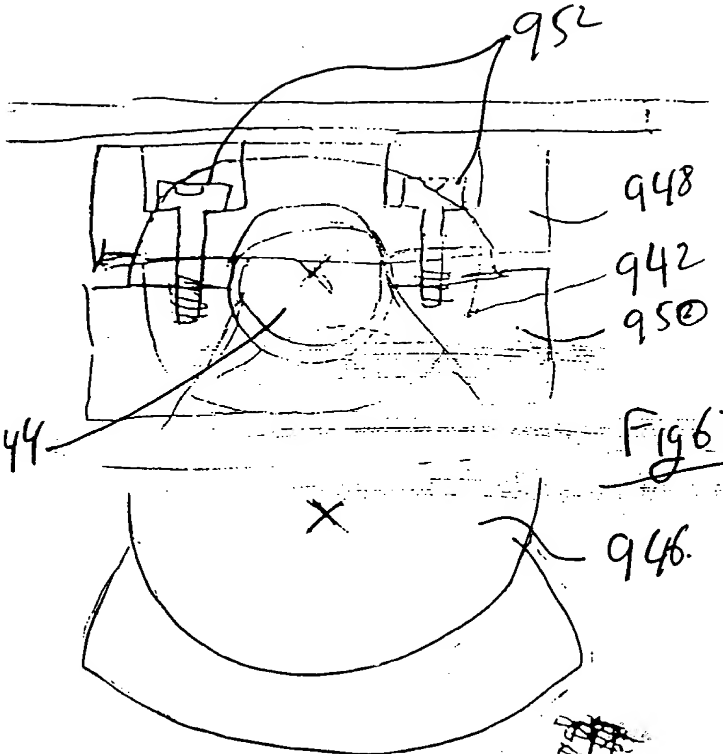

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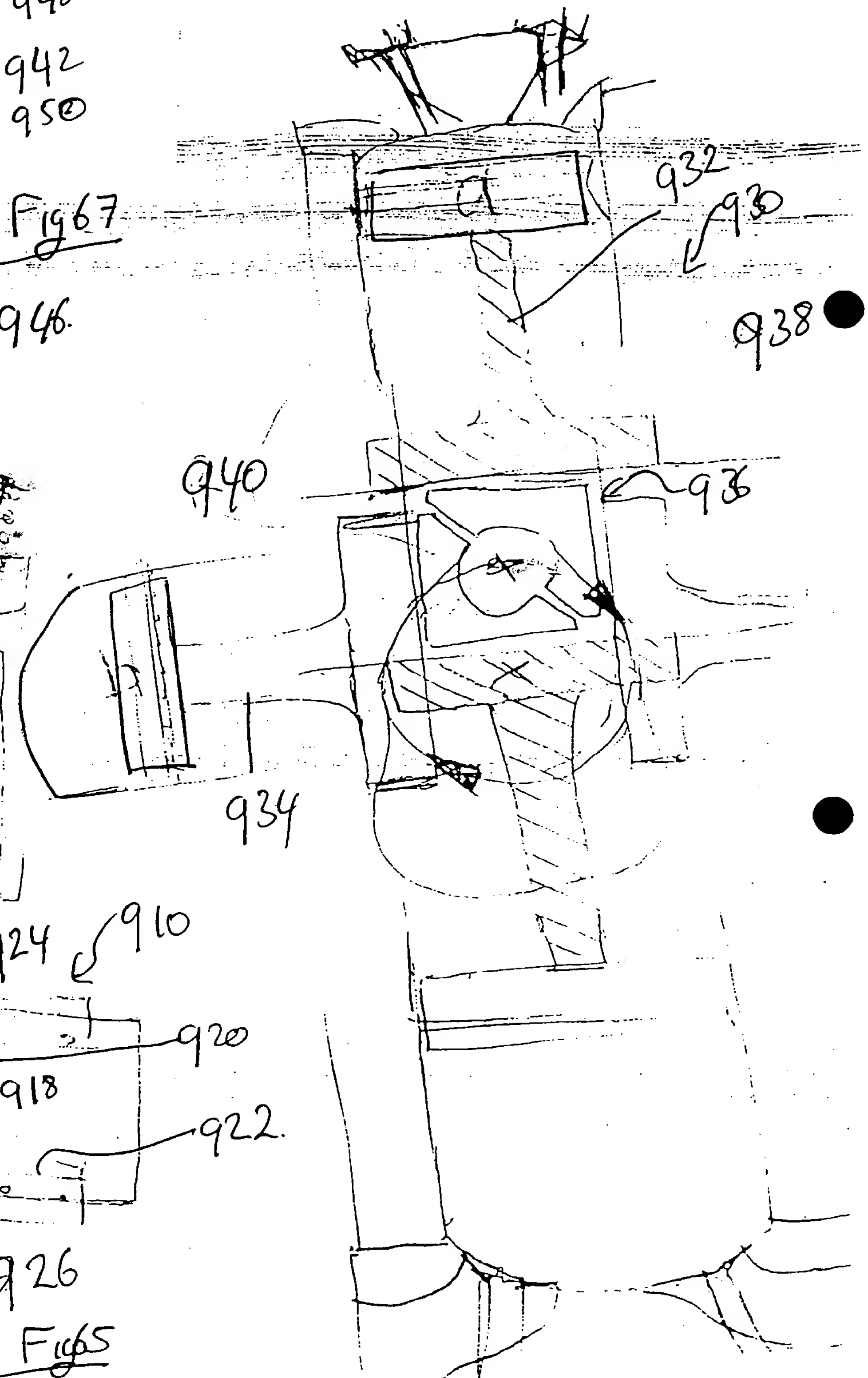
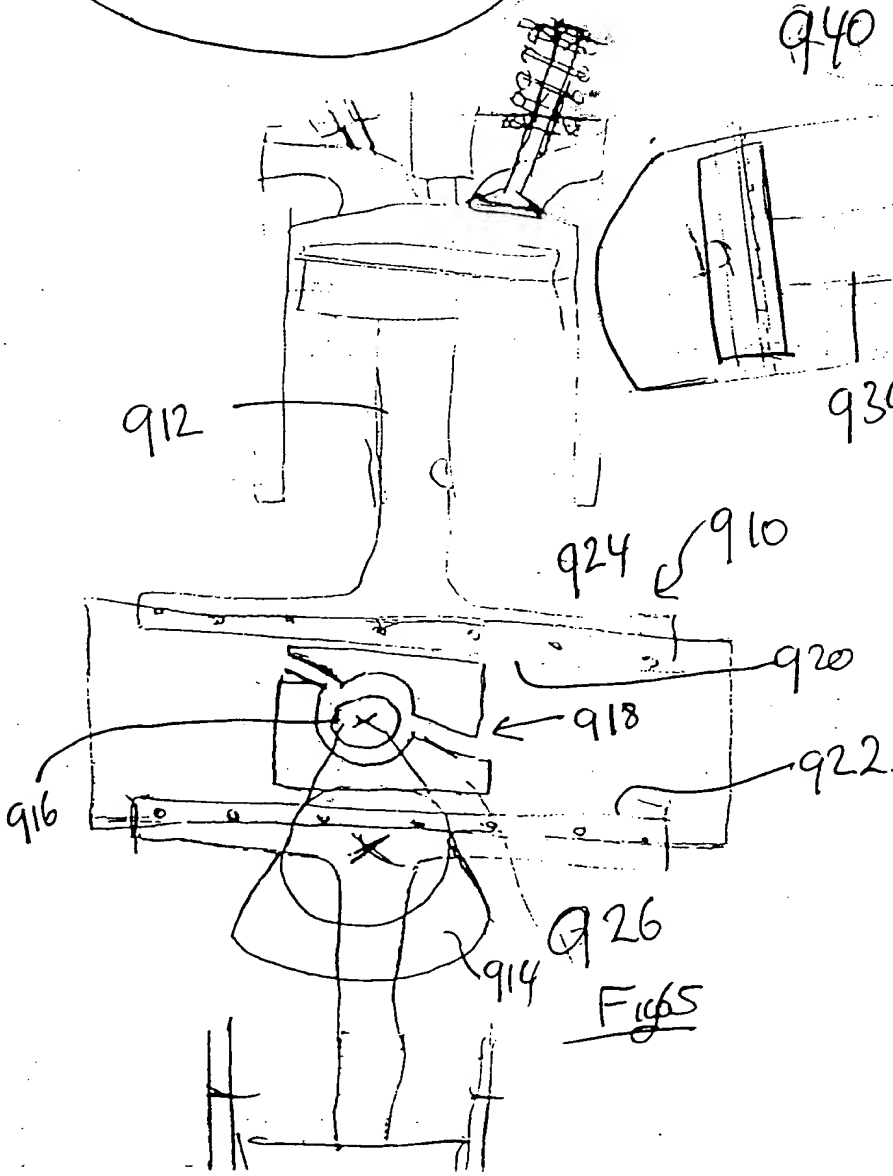


Fig 66.



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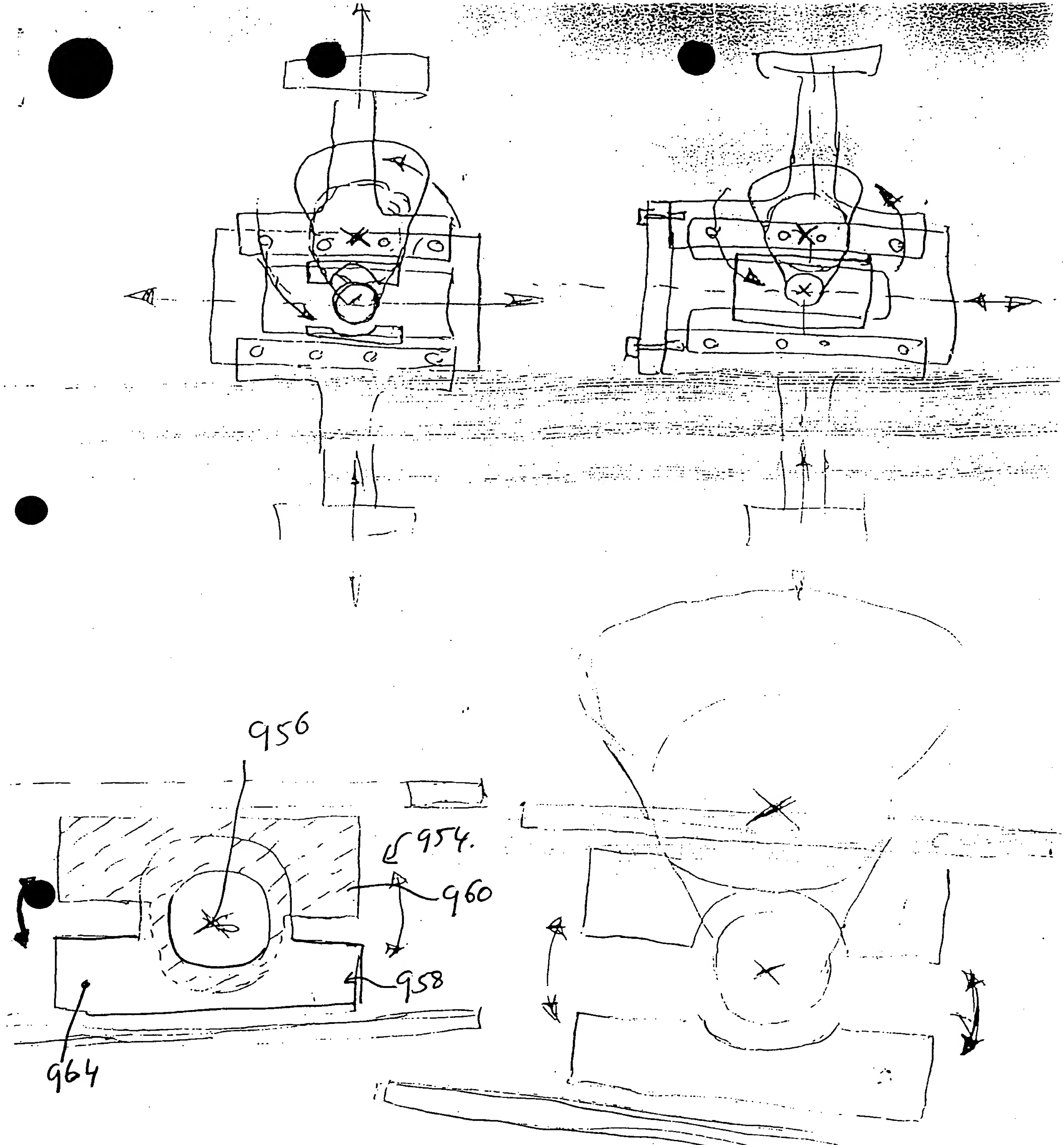


Z Fig 68

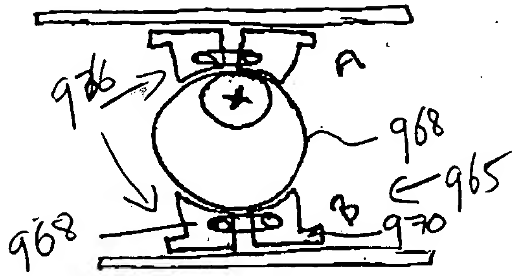


Fig 69

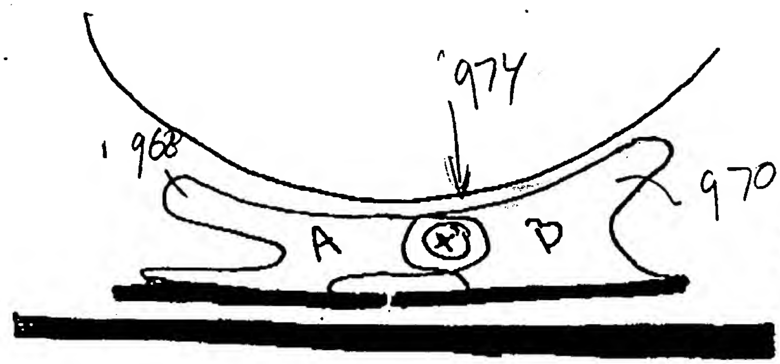
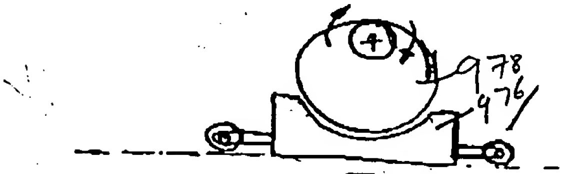


Fig 70



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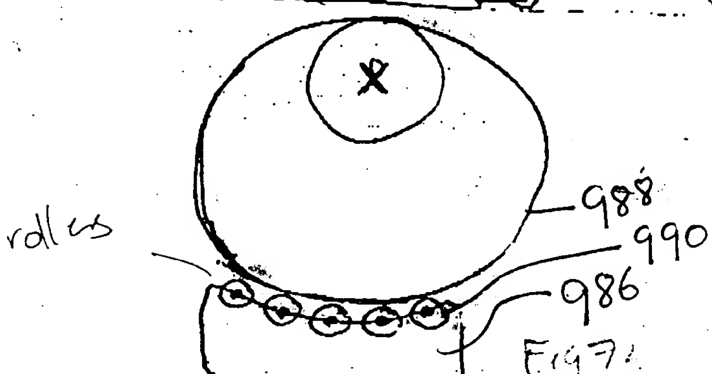

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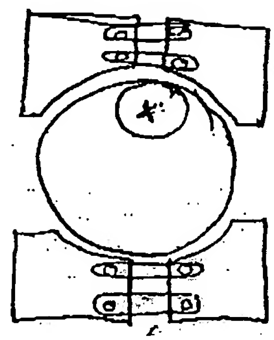

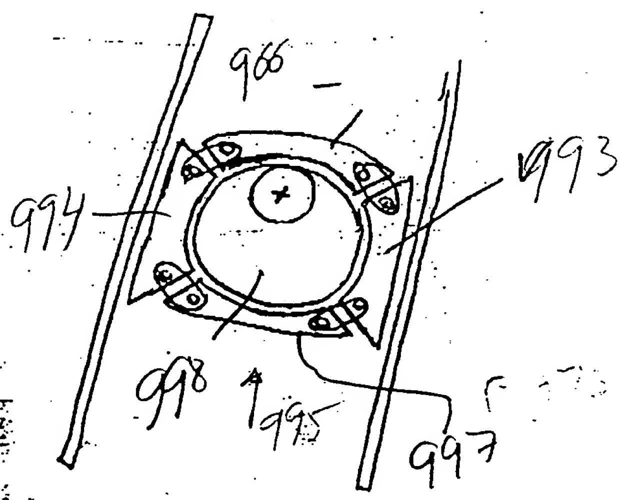

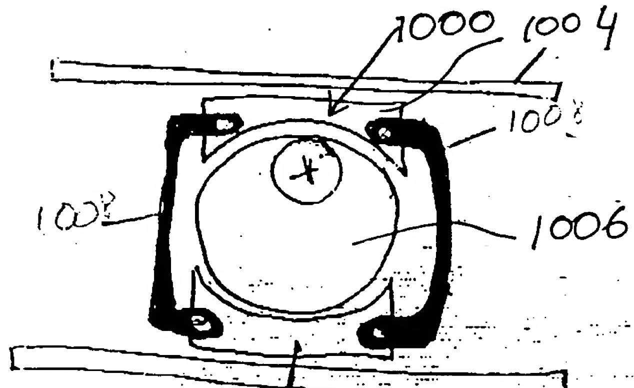

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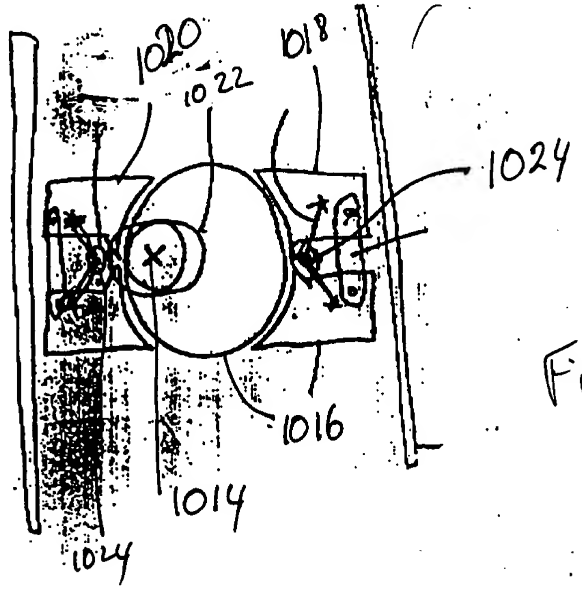

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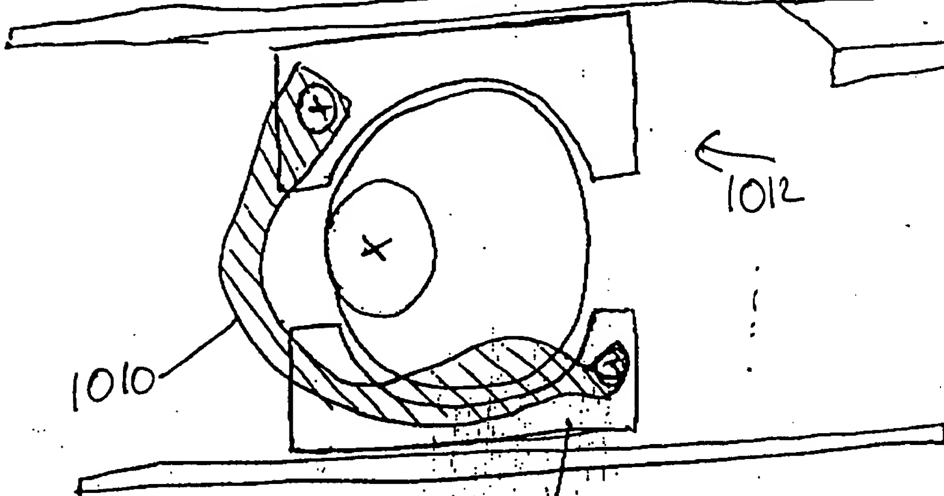

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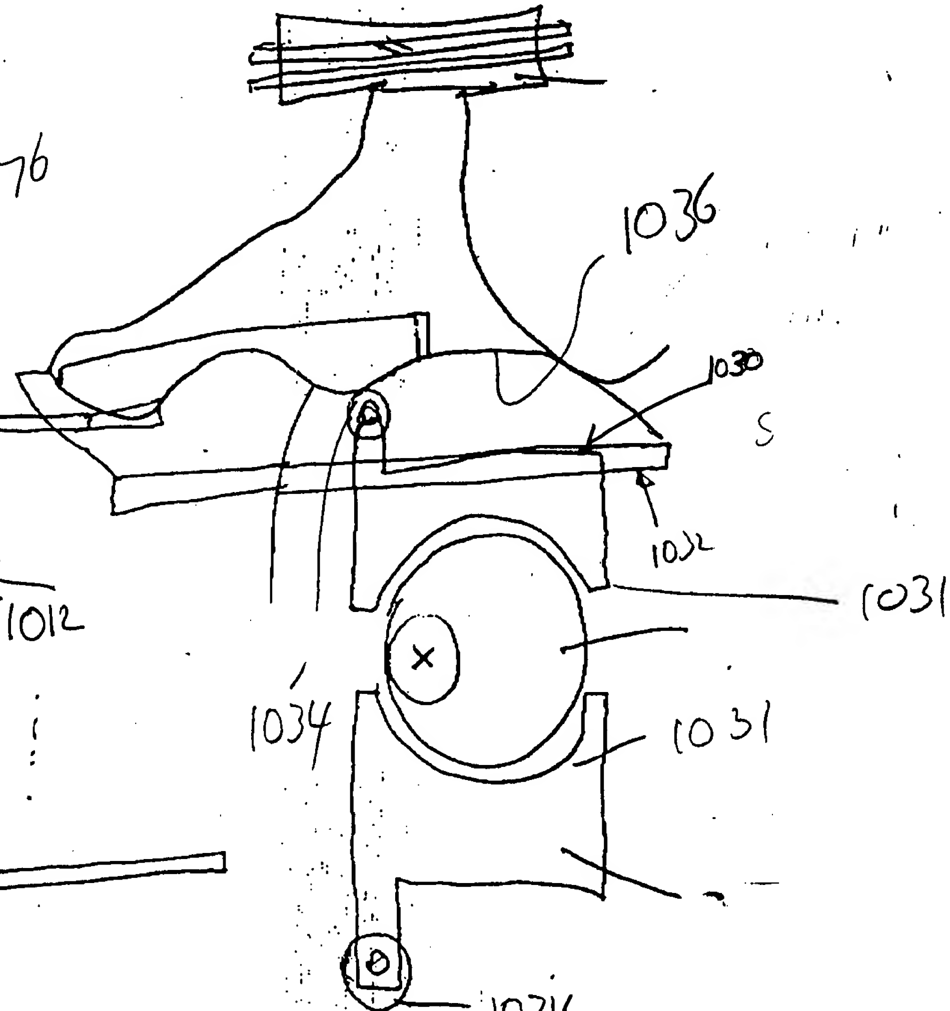

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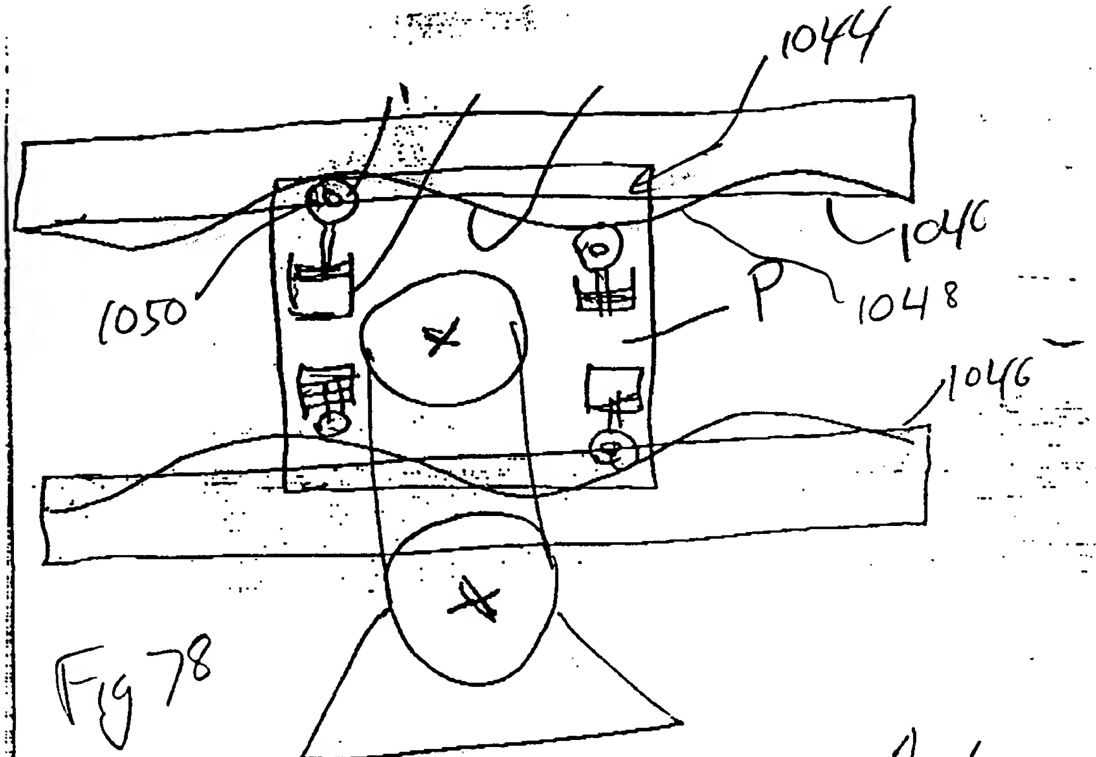

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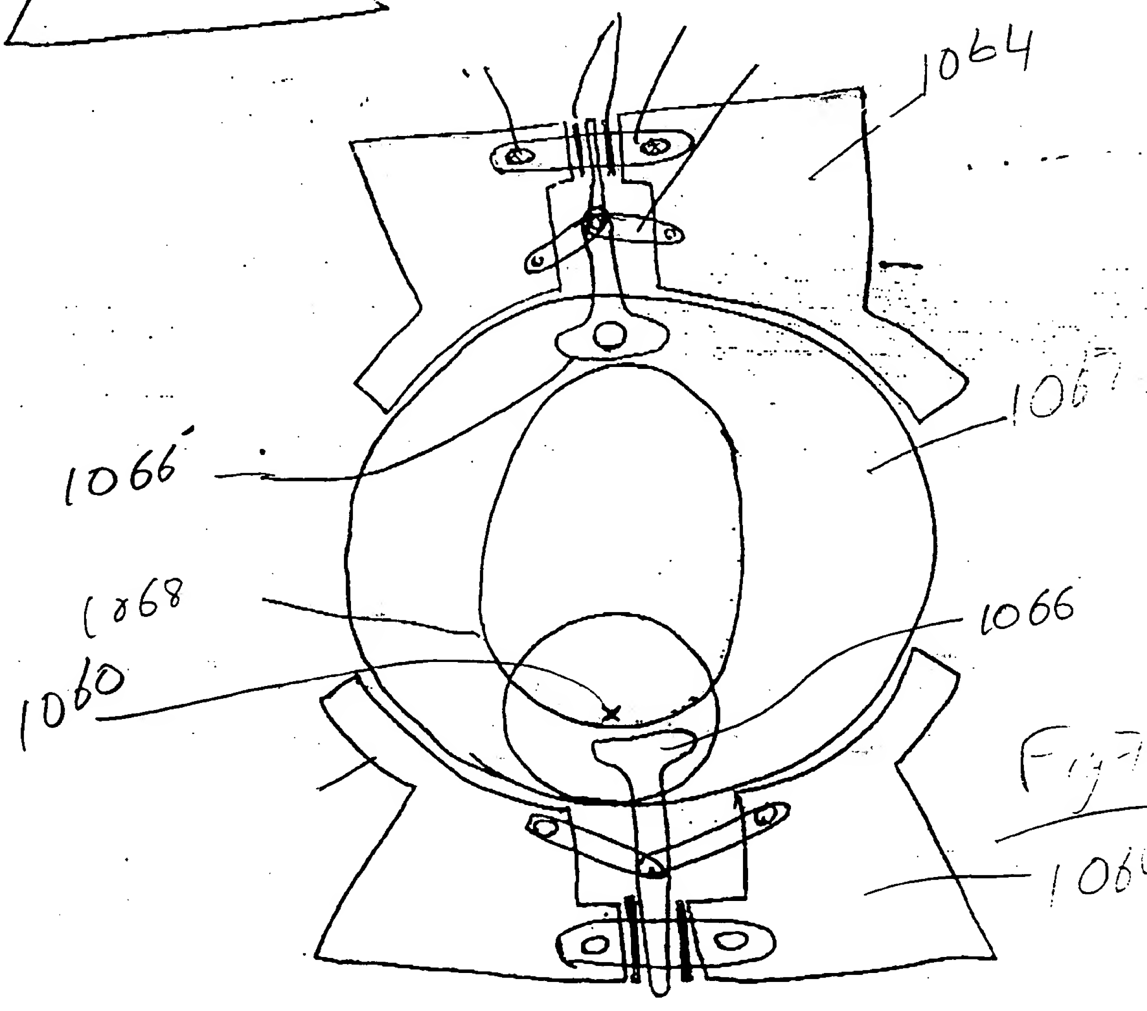

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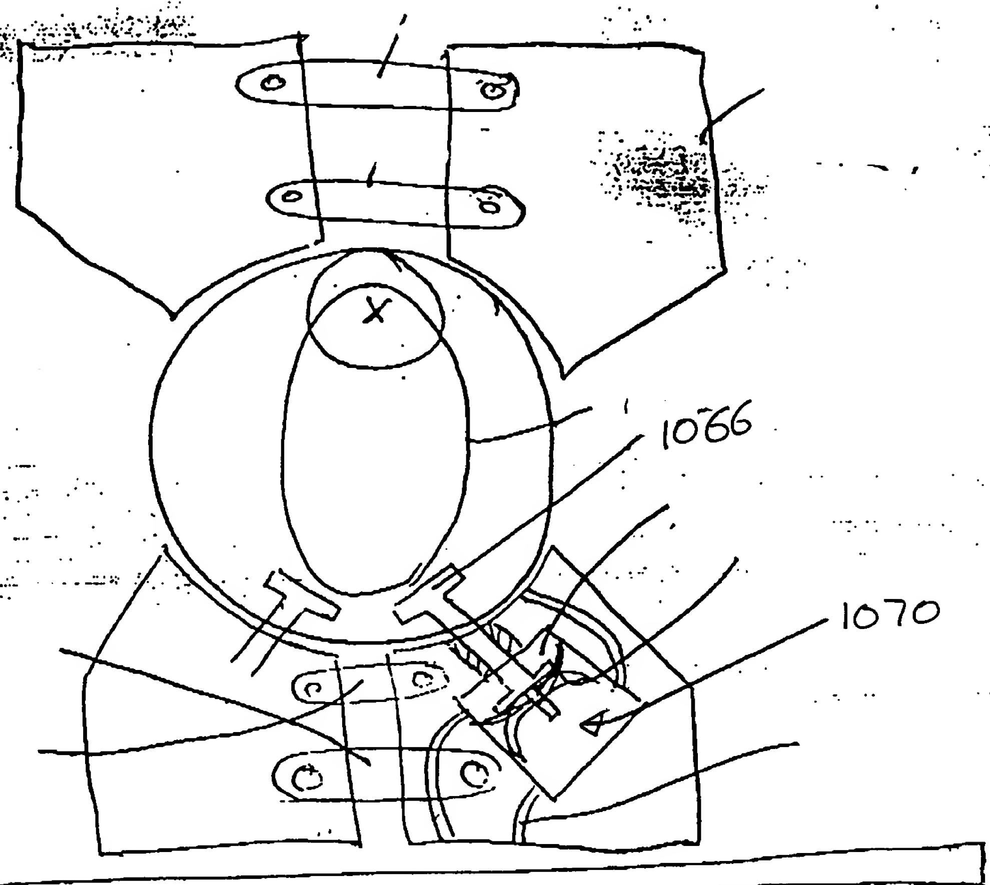

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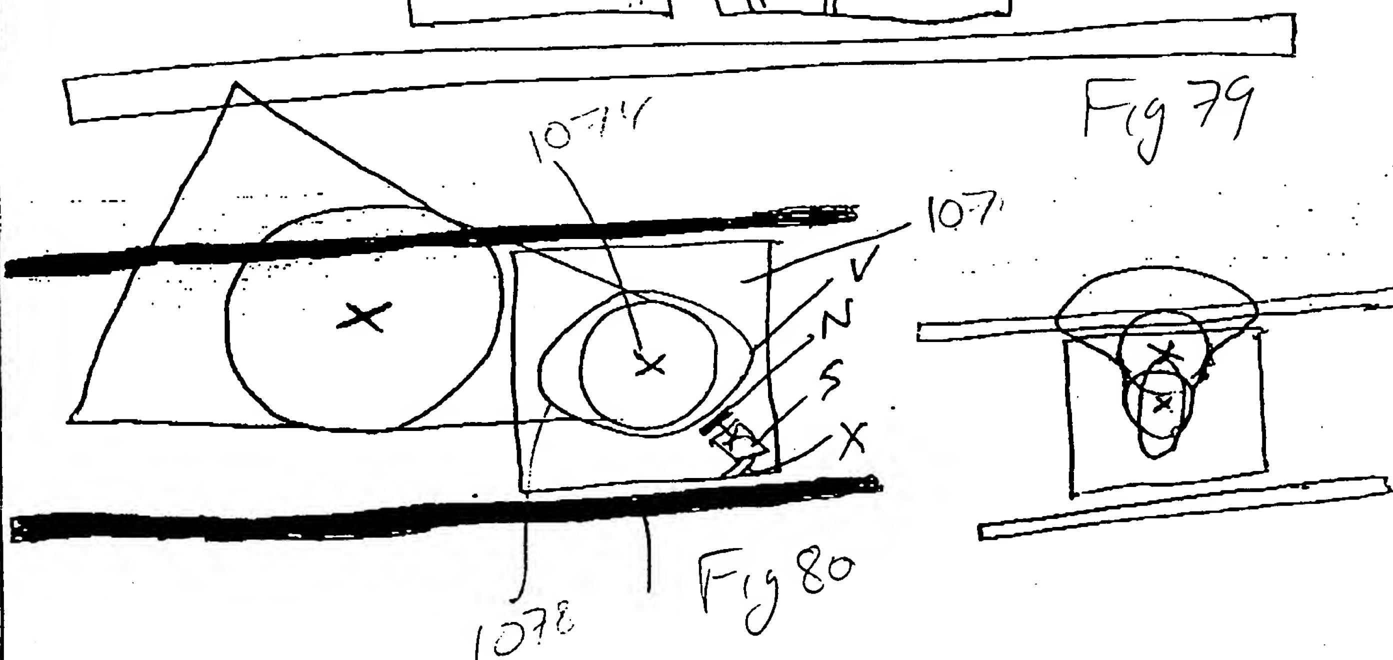

Fig 80